MOTORBOOKS WORKSHOP

Carroll Smith's **Nuts, Bolts, Fasteners and Plumbing** Handbook

- *...eristics of Metal*
- *...and Plumbing*
- *...Quick-Release Fasteners*
- *Joints, Brackets and Mounts*
- *Washers and Miscellaneous Fasteners*

Carroll Smith

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Characteristics of metal

I am going to start out with a brief discussion of a subject that scares most people to death—metallurgy. I am going to do so because I firmly believe that, once you have gone so far as to master the multiplication tables, it is no longer possible to learn without understanding. In other words, it is not enough to know which piece of hardware to use for specific jobs; there are simply too many different applications to allow anyone to master the subject by rote. In order to intelligently select, for example, the proper fastener for a given job, you must first understand how fasteners actually work. Since most of the components that you will be fastening, and almost all fasteners and other hardware are made from metal, in order to understand how fasteners work we must first understand what metals are all about—the basics of metallurgy.

Smith's law of basic simplicity

The trouble with the main body of engineering knowledge is that the people who write the books sometimes seem to go out of their way to make simple things appear complex. My feelings on the matter are, if we don't go deeply into the mathematics (and in most cases there is no need to do so) even the most esoteric of the physical sciences can be reduced to fundamentals simple enough for anyone to understand. I once heard a deservedly renowned aerodynamacist say, in response to a highly technical question, "Well, we understand that, but we cannot yet *solve for it.*" In other words, while we know basically how it works, we don't yet have enough knowledge to be able to put numbers on it. That is the type of knowledge I have tried to put forth in this book. Let somebody else write and solve the equations; I have taken the few numbers that I felt were needed from published tables.

From necessity, this chapter is going to get reasonably technical. Don't let that scare you. I promise to stay away from the mathematics.

Nature of metals

Most of us learned in high school that all matter is composed of atoms. Mother Nature has arranged for about 100 different kinds of atoms, very few of which will be of concern to us in this book. We call the different kinds of atoms elements. Each element is unique in the way in which its atoms are structured and arranged, and all of the atoms of any given element are identical. Atoms can exist in three states:

First, as pure elements, each with its own unique physical and chemical characteristics. Iron, for instance, is a soft and weak metal, while carbon is a hard and very strong but brittle nonmetal.

Second, as a mixture or a solution of different elements. In this case the characteristics of the mixture will generally be a combination of the characteristics of the constituent elements. Steel, for example, is a solution of carbon in iron, which combines some of the ductility of iron with some of the hardness and strength of carbon.

Third, as a chemical compound in which the elements of the constituents have combined in definite proportions and in a definite structure to form a new substance whose physical and/or chemical characteristics may be completely different from those of its constituent elements. As an example, sodium, a soft white metal, combines with chlorine, a green, poisonous gas, to form sodium chloride—which we know as common table (or sea) salt.

Webster defines a metal as "any of a category of electropositive elements that are usually whitish, lustrous, are able to deform plastically without breaking and which exhibit high tensile strengths combined with moderate levels of elasticity." Translated, this means that metals exhibit the following characteristics.

First, they are good conductors of both heat and electricity. For example, if you pick up a lump of metal in one hand and a lump of stone in the other, and both are cooler than your body temperature, the metal will feel colder to your touch than the stone. If, on the other hand, both are above body temperature, the metal will feel warmer.

For the moment, this ability to conduct heat (and electricity) with unusual efficiency is of no great interest to us—except, perhaps, to demonstrate the practicality of theoretical knowledge. When we are wiring a starter motor into an engine circuit, for instance, we must remember that as the temperature of a metal increases, its electrical resistance also increases and therefore its ability to conduct electricity decreases. This is why we need large wires from the battery to the starter. It takes a lot of current to crank the engine over. The transmission

of this current through the starter cables heats the wires; the heating of the wires decreases their ability to conduct current so the starter motor turns the engine over more slowly and, if the wires are too small, the engine won't start. This example will serve as our first practical application of metallurgical knowledge.

A second characteristic is that metals are lustrous. This means that they reflect light well and/or that they shine well—a characteristic of interest only when we find ourselves polishing something, in which case we usually wish that it weren't so.

Third, metals as a family are, to some extent, plastic. They are generally able to stretch a long way before they break. This is one of the key factors that makes metals useful to us. It is this characteristic that allows metals to be bent, stretched, beaten or rolled into sheets, bars and rods, drawn into wire and formed into useful shapes.

Fourth, metals are elastic. When a metal shape has been deformed by an applied load, it will return to its original shape and size when the load is released—assuming that the level to which the shape was loaded was within the elastic limits of the metal. It is this elasticity that allows metal structures to resist failure under high levels of load and stress. A correctly designed and fabricated metal structure can survive severe localized and impact loads by giving or stretching slightly in the area of the impact, thus distributing the load throughout the structure. Under the same conditions, a ceramic or reinforced plastic structure, lacking the elasticity of metals, would be liable to crack and/or fracture in a brittle manner.

A fifth characteristic of metals is that they are strong. Without strength, metals would be useless as structural materials. Fortunately the creator foresaw this and endowed metals with their characteristically high tensile strengths, making them the most useful and versatile of all groups of elements.

Properties of metals

Before we go any further into our investigation of metals, we must pause and define some of the terms that will be used to describe their properties.

Load is the overall force to which a material or a structure is submitted in supporting the weight of a mass—or in resisting an externally applied force. In the United States, load is usually expressed in pounds.

Stress is an applied force that tends to deform a body or a structure. It is usually expressed as unit stress, the load divided by the area over which it is applied. In this book we will express unit stress in pounds per square inch (psi).

Strain is the term used to describe the change in dimension of a body or structure as a result of stress.

Strength is the ability of a material to resist the stress caused by applied forces. The ultimate strength of a metal is determined in a testing laboratory and is a measure of the maximum stress that can be applied to a material before it ruptures. There is no law that says that a material should have the same ultimate strengths in tension, compression and shear. In fact, they do not. The strength of cast iron in tension, for example, is only about one fourth of its strength in compression. The ultimate shear strength of alloy steels is about one half of the alloy's ultimate tensile strength.

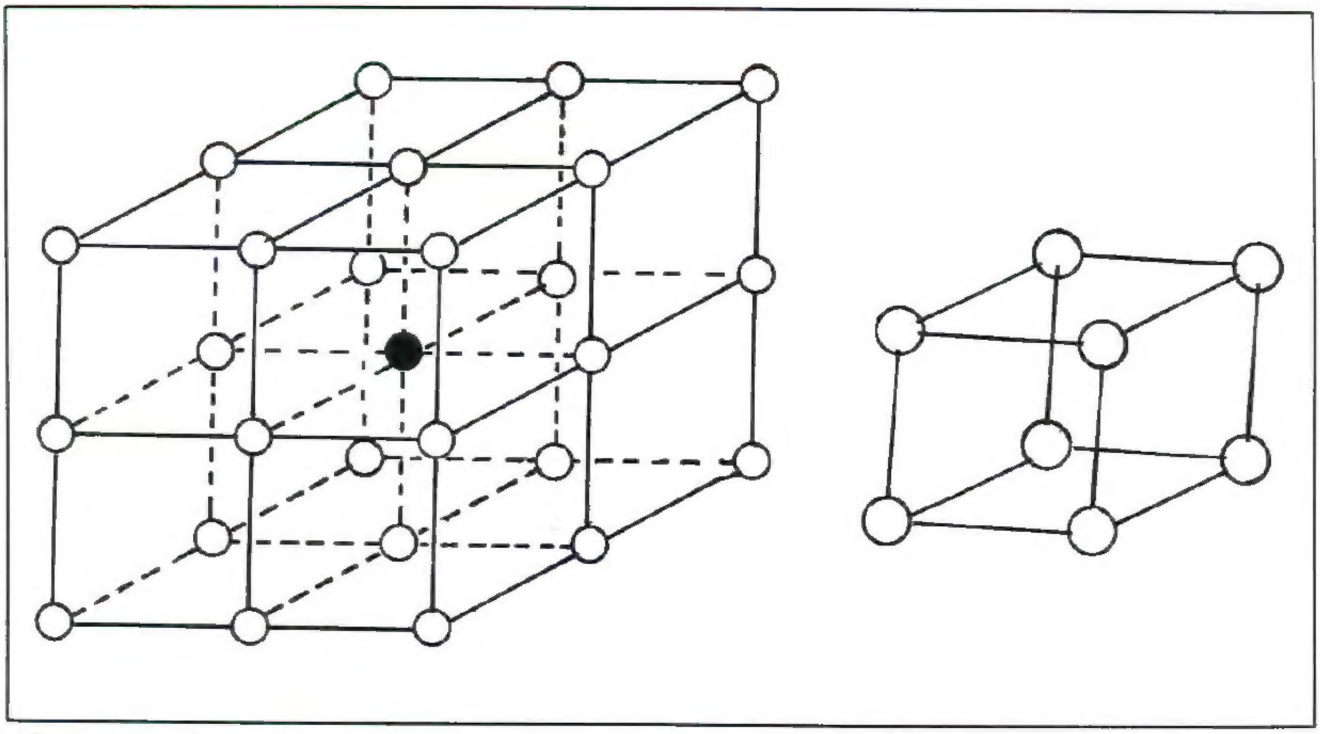
Hardness is the property of resisting penetration or wear. Normally, the hardness of a material varies directly with its strength—the harder the material the stronger it will be, and vice versa. Examples of hard materials include glass, high alloy steels, tusks, teeth and most gemstones. The opposite of hard is, of course, soft. Soft materials include lead, most woods and the weaker grades of aluminum.

The toughness of a material is defined as the total amount of energy that the material can absorb before failure—a measure of the material's ability to resist impact or shock loads. It is the opposite of brittleness. Examples of tough materials are most metals, bone and hard woods.

Brittleness is the tendency of a material to fracture without changing shape. Technically, brittleness is the property of resisting any attempt to change the relative positions of the atoms within the structure of a material. Unfortunately, hardness and brittleness enjoy a close relationship—the harder (and therefore the stronger) a material is, the more brittle it is liable to be. However, just because a material is brittle does not mean that it is either hard or strong. A Hershey bar, for instance, is obviously both soft and weak—but it is also brittle. Try bending one. Brittle materials exhibit poor resistance to impact and shock loads and are therefore not generally suited to engineering use.

Malleability is the exact opposite of brittleness—a measure of the ability of a metal to accept permanent deformation in all directions without rupture. In order for a metal to be shaped or worked by rolling, pressing, hammering, drawing and so on, it must be, to some extent at least, malleable. Many metals (such as steel) are malleable only when heated. A malleable material can be severely bent (the technical term is permanently distorted) without rupture. Examples of malleable materials include aluminum, copper, low-carbon steels and pastry dough. As a point of interest, the most malleable of the metals is gold. Unfortunately, malleable materials, at least those that are malleable at normal temperatures, tend to be weak.

Ductility is quite similar to malleability. It is the property of some materials that allows them to be drawn out to thin sections without breaking, like taffy or wire. In order for a metal to be ductile, it must be strong. We often see the term ductility misused to mean both ductility and malleability. This is a mistake. Just because a metal is malleable



A hypothetical simple cubic unit cell and a cubic crystal space lattice composed of eight cells. Notice that, with the exception of those located at the outside corners of the lattice, each atom is shared by at least two unit cells.

does not mean that it is ductile. Lead, for instance, while it is exceptionally malleable, exhibits little ductility. Examples of ductile materials are aluminum, copper, silver and gold. Nonductile materials include alloy steels, glass (at normal temperatures) and wood.

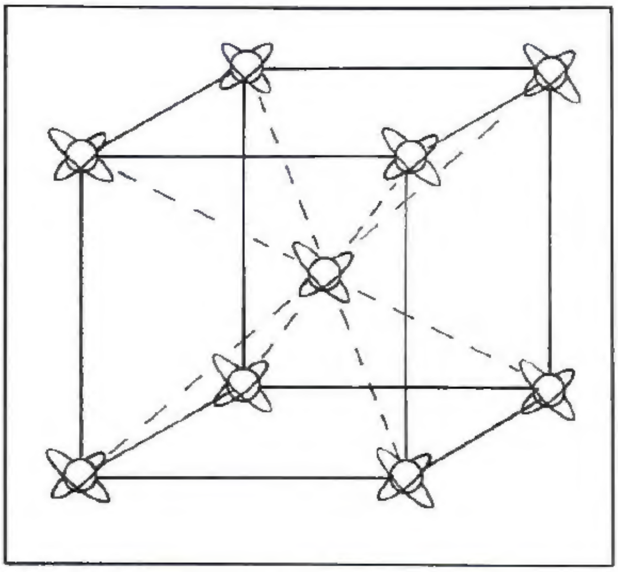
Elasticity is a measure of the springiness or the stiffness of a material. Technically speaking, elasticity is the ability of a material (or a structure) to deform elastically in response to a force or stress and then to return to its original shape when the force or stress is removed. Elasticity is the opposite of flexibility. Examples of elastic materials are glass, most metals and rubber.

We have looked at the general nature of metals, investigated some of their characteristics and defined some of the basic terms. It is time to begin our more detailed investigation into their structure and behavior.

Crystalline structure of metals

Although they certainly look and feel like solid entities, all metals are actually polycrystalline in structure; that is, they are made up of joined groups of separate crystals. A crystal is defined as an orderly and repetitive arrangement of identical unit cells that are in turn each composed of a fixed number of atoms arranged in an unvarying pattern in space.

By their very nature, crystals do us some favors. They are organized little devils. The arrangement of atoms within any given crystal cell is orderly, predictable and repetitive. Each atom vibrates or does its thing around a fixed point and these points are cleverly arranged in a regular pattern in three-dimensional space. What is more, the pattern repeats itself so that the individual cells can join each other in an orderly fashion in a modular three-



Nine atoms of copper forming a cubic crystal cell of metallic copper.

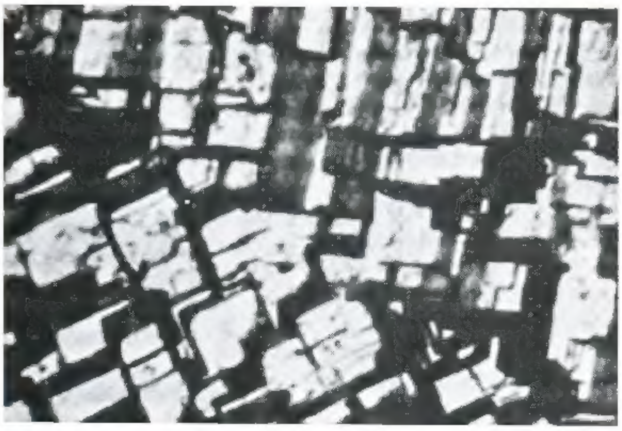
dimensional "crystalline space lattice" which resembles a LEGO toy set.

Crystal unit cells come in several configurations. We are not going to concern ourselves with that, however. For our demonstration purposes, we are going to consider them to be cubes, for a couple of reasons. One is that a great many metals, including iron and aluminum, do exhibit cubic crystal cells. The other reason is that they are easy for me to draw.

Crystalline lattice

Visualizing in three dimensions, let's take a look at the construction of the idealized cubic crystal lattice. Notice that, with the exceptions of those located at the outside corners of the lattice, the atoms at the corners and edges of each cell are shared by the other cells having the same corners and/or edges, and that the atom located at the center of the space lattice is actually common to all of the unit cells shown. Although each module or unit cell of the space lattice can be rotated every which way about its various axes, when the rotation is stopped, it will still fit perfectly into the crystal lattice—rather like square LEGOs. Each of the crystal unit cells can be refitted into its proper position within the lattice by a rotation about any of its axes of rotation. It will fit perfectly, sharing edges and atoms in a unitary structure.

As an indication of the numbers and sizes of the particles involved, let's consider the structure of one of the more common metals, copper. Each unit cell of copper is composed of nine atoms arranged in the shape of a cube, with one atom at each corner and one in the geometric center of the cube. One gram of pure copper contains 2,370,000,000,000,000,000,000 unit cells or 21,330,000,000,000,000,000,000 atoms of copper. To put things into perspective, the smallest object that can be distinguished by the unaided human eye, under ideal conditions, is about 500,000 atoms in diameter.



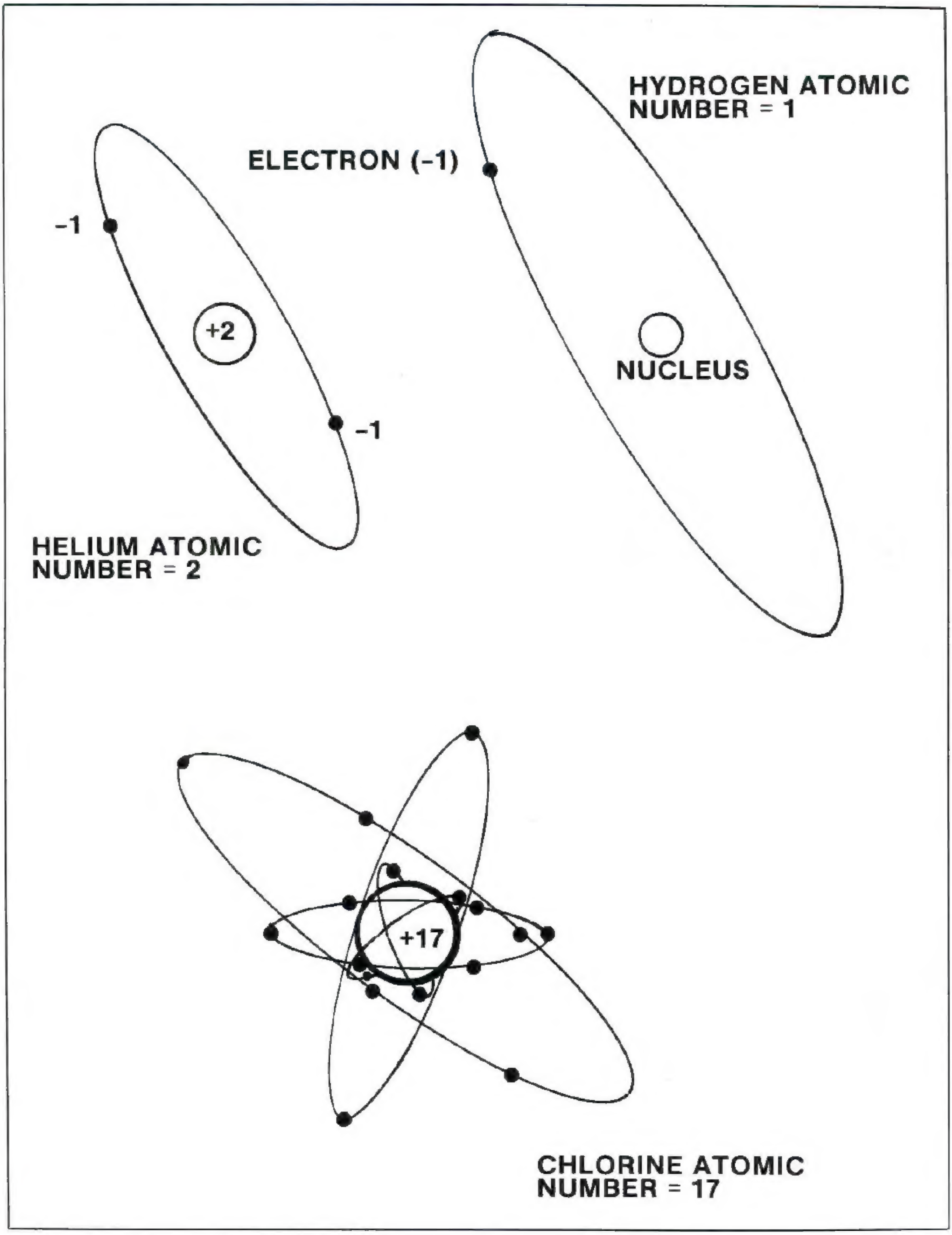
Grains of aluminum enlarged 5,000 times.

The basic structure of the crystals—the actual arrangement of the atoms into unit cells—is, logically enough, termed the atomic structure of the metal. Since the individual crystals (which we usually call grains) are normally far too small for us to see, they are examined through a microscope at magnifications ranging from 100 to 1,000 times and more. Crystal structures with grains small enough to require this sort of treatment are referred to as microstructures. Some castings have structures so coarse that individual grains can be distinguished with the naked eye. These are called macrostructures and are of little interest to us, thus I will not mention the word again.

This is probably as good a time as any to state that there is no such thing as a molecule of metal. Metals have atoms, unit cells and crystals, but they do not have molecules. In fact, one of the characteristics that distinguishes the family of metals from other elements is that they are monotomic—that is, their crystalline structure is made up of individual atoms and not of molecules.

My trouble in this area lies in my difficulty in visualizing the atoms themselves. After all, no one (at least no one that I know) has ever seen an atom! As a long-time science fiction fan I have no trouble at all visualizing the planets whirling in orbit around the sun and the moon orbiting earth, and the other planets orbiting the sun, each with their own satellites, while the whole mess—solar systems, comets, asteroids, Han Solo, Princess Leia, E.T. and all—rushes through space. I have a hell of a lot of trouble looking at my own fingernail, though, and realizing that this seemingly solid object is mainly empty space. But empty space it surely is, simply because it is composed of countless atoms, each of which is in turn comprised of a dense nucleus surrounded by a cloud of electrons, each of which is whirling in its own little orbit around the nucleus. The space between the nucleus and the orbit of the electrons, which makes up the vast majority of all space, is empty. My brain knows that this is true, but my heart (or some other nonreasoning part of me) is boggled by the concept.

Regardless of the configuration or atomic structure, atoms are held together by strong electrical forces. The nucleus is composed of positively charged protons and neutrally charged neutrons, while the orbiting electrons are negatively charged. To the best of my knowledge, no one knows exactly what an electron really is—they are called waves, particles and negative charges of electricity and, lately, quarks. We are going to consider the electron to be an infinitesimally small particle of matter that has a negative electrical charge. We are not going to worry about how either the electron or the charge got there. Opposites attract and so the whole system is kept in equilibrium by electricity—similar to the way in which gravity and velocity interact to keep satellites in earth orbit.



The atomic structure of some simple atoms.

So, to sum up what we've discussed thus far, in the real world, metals are a collection of atoms that fly around in very close formation and are arranged in a regular and repetitive fashion into crystal unit cells of various shapes. These shapes are, in turn, built up, LEGO-like, on a regular and repetitive three-dimensional lattice structure into crystals of metal which we call grains. Metals are ductile, strong and elastic. These characteristics allow us to form metals into shapes that will resist loads and impacts.

The next question is, of course, How do the metals manage to do all that? The answers are not simple, nor, at the time of writing, are they entirely understood—at least by practical engineers like me. A basic idea of what is actually happening when we form metals will, however, go a long way toward preventing mistakes in their use and consequent structural failures. So bear with me.

Plastic deformation of metal

One of the major characteristics that makes the family of metals more useful to engineers than, say the family of minerals, is their unique ability to undergo large permanent deformations (plastic deformation) without rupture and still remain strong and elastic. In other words, we can form the damned things into shapes that—after they are formed—will be strong, tough and resistant to both shock and fatigue. This seeming magic comes about because the crystal structure of metals, unlike that of minerals, is relatively tolerant of structural imperfections. Metals form strong but flexible bonds across the boundaries between their adjacent grains. Nonmetallic crystals may form strong bonds across grain boundaries, but they do not form flexible ones and so are liable to rupture across grain boundaries whenever stress is imposed—as in splitting a diamond. In order to explain why this is so I must touch, briefly, on the structure of the atom itself and on the nature of the interatomic bonds that hold our universe together as well as everything and everybody in it.

Structure of the atom

We are all more or less familiar with the concept that everything and everybody is a collection of tiny atoms flying around in close formation. We also know that each atom consists of a nucleus around which orbit a number of electrons—much as the moon orbits the earth and the planets orbit the sun. The nucleus is comprised of some number of protons each of which has a positive electrical (+1) charge and some number of neutrons, which have no charge. The orbiting electrons each have a negative electrical charge (–1). The mass of the nucleus is many thousands of times greater than that of the electrons, but the space enclosed within the electron orbits is many thousands of times the volume of the nucleus.

We know that in physics, just as in human relationships, opposites attract. In each case the attraction has to do with electricity, sexual in one case and physical in the other. We also know that the electrostatic attraction between positively charged protons and negatively charged electrons is what holds the universe and everything in it together. Atoms of different elements have different numbers of neutrons, protons and electrons and, consequently, different weights. The atomic number of any element is simply the number of protons in its nucleus.

For example, the most simple of all atoms, hydrogen, has exactly one proton and a corresponding atomic number of one. The hydrogen nucleus has a positive electrical charge of +1. Orbiting this simple nucleus is a single electron with a negative electrical charge of –1. Overall, every atom is electrically neutral. More complex (or heavier) atoms have more protons, more neutrons and more electrons to match their protons. As examples, sodium has eleven protons and an atomic number of eleven, chlorine has seventeen, iron twenty-six, uranium ninety-two and so on.

This picture that I have painted of the atom as a dense nucleus surrounded by tiny orbiting electrons is accurate—the nucleus is typically 1/10,000 to 1/100,000 of the diameter of the outermost electron orbit. It is so dense that one cubic centimeter of tightly packed nuclei would weigh 220,000,000,000 pounds.

Accurate my picture may be—but it is incomplete. Somewhat like the earth satellites with which we are becoming familiar, there are several possible electron orbital distances and inclinations. In fact, like the satellites, the orbital distance from the nucleus depends on the amount of energy with which the electron in question is charged. Unlike the earth satellite, however, there is a finite and fixed number of orbits available to the electrons. And rather than being infinitely variable in diameter, they can vary only in finite and fixed steps. Further, the maximum number of electrons that can fit into any given orbit is limited. As you would expect, the further the orbit from the nucleus, the greater is the amount of energy required to maintain the electron in orbit—just as you would expend more energy whirling a rock about your body on the end of a long string than you would whirling the same rock at the same rpm on a short string.

Mother Nature is not only intelligent, she is also lazy. One of the irrevocable laws of nature is that the higher the total energy contained in any physical system, the less stable the system will be. Being lazy, nature always tries to achieve the lowest available energy state, with all components as close to a state of rest as can be arranged. For this reason, in the structure of any atom the possible electron orbits closest to the nucleus are filled first, and electrons select the higher orbits only after the

lower ones have been filled. The electrons in the outermost orbit of any atom are highly charged and relatively unstable little devils.

Atomic bond

Something must hold together all of the atoms of any given element, alloy or compound. We have seen that the electrostatic attraction between the positively charged protons in the nucleus and the negatively charged electrons in orbit about it holds the atom together. But each atom is, by definition, electrically neutral and so, since there is no attraction between neuters, it would seem that electrostatic attraction cannot bond individual atoms together into the clumps that make up matter. Wrong! It is indeed the old attraction between opposites that not only binds our world together, but also applies in methods more complex than the simple plus-minus relationship that we have seen in the case of the individual atom.

Strangely enough, the complex and dense nucleus of the atom is pretty much a fixed entity—both in makeup and in charge. The exception is nuclear fission, without which we would all be better off. The neutrons and even the protons don't have much to do with the formation of interatomic bonds, except of course that the protons do provide the positive charge without which nothing would work. (It's a little bit like the male role in conception and childbirth.) On the other hand, the busy little electrons in the process of whirring about in their chosen orbits can and do interact with the electrons and orbits of other nearby atoms. They can interweave orbits and, being faithless little devils, they can desert their own nucleus and either go off with another, share a common nucleus, or even share their affections and attentions among a whole group of nuclei. In so doing they form interatomic bonds of varying flexibilities, strengths and longevities. There are several categories of atomic bonds; we are interested only in the metallic bond.

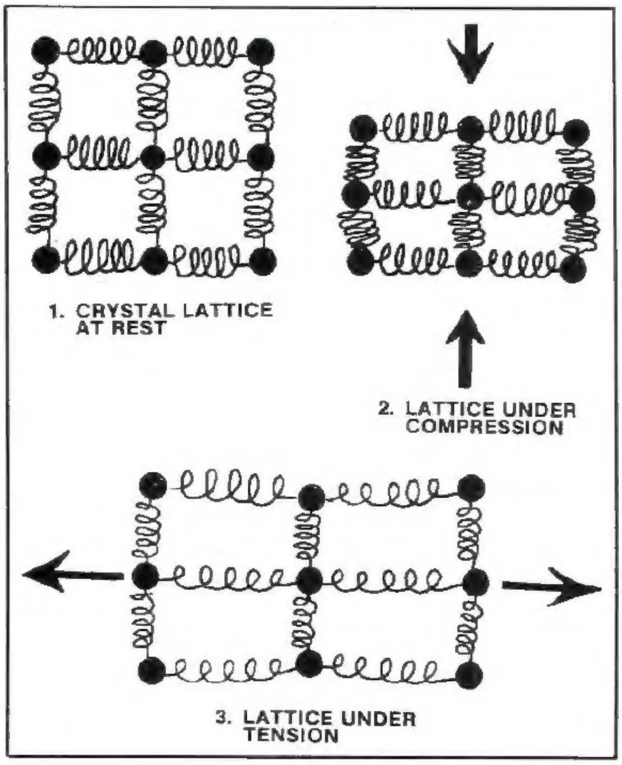
Metallic bond

The bond between atoms of metal is unique to the family of metals. It is the characteristics of this bond that impart to metals their unique combination of strength, elasticity and ductility. When the atoms of any given metal combine or link to form crystals, a close investigation will reveal that within the crystal, the outer orbital shell of each atom is missing a few electrons. Two questions immediately arise: Where are the missing electrons and, since all of the atoms are now positively charged, and likes repell, why doesn't the crystal fly apart?

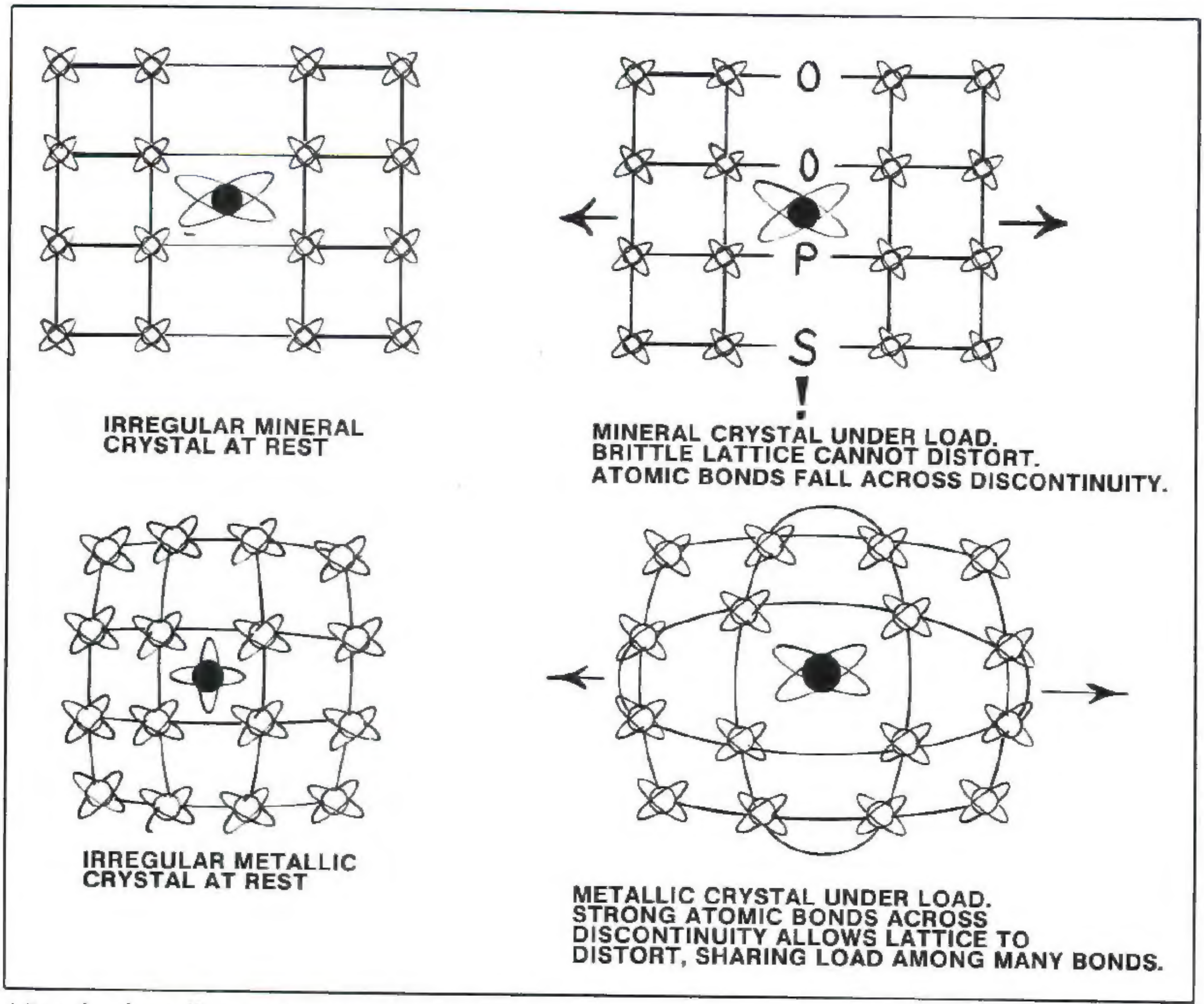
The answer to the first question supplies the key. The detached electrons are, in fact, still present (and busy) within the crystal. One of the characteristics of metallic crystals is that the atoms (or, more accurately, the ions) are packed closely together in space. This somewhat alters the rules of the game. The extremely close packing of the nuclei causes a change in the energy relationships. Under these conditions, the lowest energy level for the outermost (highest energy and most unstable in the orbital sense) electrons is no longer a fixed orbit around a single nucleus. Instead it becomes a sort of wandering, random orbit or series of orbits throughout the entire crystal.

At any given time there are enough electrons—those in fixed orbit plus sufficient wanderers—*in the vicinity of each nucleus* to maintain both individual and collective neutrality of electrical charge. The particles that would seem to be positively charged are, in effect, neutrally charged atoms of the metal. At the same time, the electrons in random circulation or orbit form the flexible adhesive that bonds the atoms together, with very strong but flexible interatomic bonds, into a crystalline lattice which is remarkably tolerant of structural imperfections. These interwoven random orbits, by their circulating nature, allow any disturbing force to be spread over all of the bonds and so resist stress that might otherwise flex the lattice structure sufficiently to break the atomic bonds and so fracture the lattice. Moreover, any local disturbances to the circulation of the electrons throughout the crystal structure tend to be almost self-healing.

These semi-free electrons are simply not present in the structure of minerals, and so cannot



Schematic representations of the atomic bonds of metals.



Mineral and metallic crystals at rest and under load.

come galloping to the rescue when they are needed to reinforce the bonds across those specific grain boundaries that are endangered at any given instant. It is important to realize that only a few electrons from each atom are ranging freely throughout the crystal—most of them are still contained in their regular orbits about their very own nuclei. It may help to think of the roving electrons as a sort of elastic mortar bonding the bricks of a wall together, but an elastic mortar that is in constant circulation among the bricks and is temporarily attached to each.

A digression

As a point of interest, for the past several pages we have been looking into the dreaded field of quantum mechanics. Admittedly our look has been both incomplete and oversimplified; hopefully it has been understandable. If so, it should serve not only to acquaint you with some of the basics of the atomic structure and the bonding of metals, but also to partially dispel the clouds of fear that tend to surround most bodies of scientific knowledge. If all scientific writers and teachers could, or would, write and teach with the clarity and simplicity of Isaac Asimov, we would all be better off and our nation would not be behind in any portion of the technology race.

At most levels, physical science is, if not simple, at least understandable in principle. We are merely afraid of it—just as I was scared to death of computers until Dave Head forced me to spend a couple of hours on his. Now I cannot believe that I ever tried to write, let alone design, without one. I truly believe that the normal human being can learn anything that he sets his (or her) mind to. Our problem is a combination of laziness and lack of self-confidence. Pity!

Elasticity of metals

We have seen that it is the plasticity of metals that allows us to form metals and their alloys into useful shapes. This is all very convenient, but if metals were completely plastic in nature then our metal parts, no matter how convenient their shape, would lack the stiffness necessary for us to make structures capable of resisting the loads that we wish to impose upon them. Fear not. All has been arranged by the master engineer in the sky.

The study of elasticity is simply the study of the relationships between stresses and strains in solid materials. In essence, the study of elasticity is the study of stiffness.

In the learning process, the instructor must define the terms that will be used before the instructee begins the study of any body of knowledge. Intuition, to the engineer, is instinct tempered and matured by experience. Much of what follows you already know by intuition. In order for us to raise the results of our engineering above the level represented by instinct, we must first formalize some of the concepts that you may think you already understand. Since no one likes definitions (they are just as boring to write as they are to memorize), I usually try to hide them within the text. Sometimes I fail, however—it is now time for a few more undisguised definitions.

To the engineer, load is the applied force to which a shape or a structure is subjected in supporting a mass or in resisting an externally applied force. We can all relate to that.

To the nonengineer, however, stress and strain are frightening words. A century ago, Sigmund Freud and his disciples, realizing that engineering terminology was both precise and convenient, applied the language of structural engineering to the distressed mental states of mankind. Since none of us are comfortable with the notion of mental disorder, the theft of our terms by the shrinks has rendered the words scary.

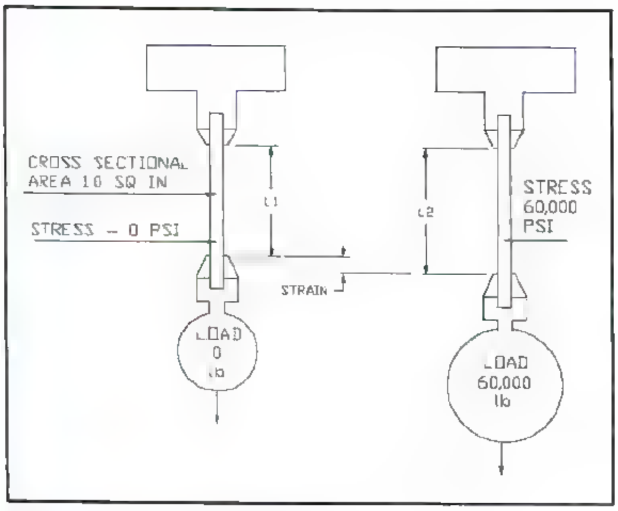
In the discussion of loads and strengths, stress is simply the applied load divided by the cross-sectional area of the loaded part. If a bar with a cross-sectional area of two square inches is supporting a load of 1,000 pounds in tension, then the tensile stress within the bar is 500 pounds per square inch.

Strain is merely the deformation produced (in this case an elongation in a member) by the application of a stress. There is nothing complex or scary in that.

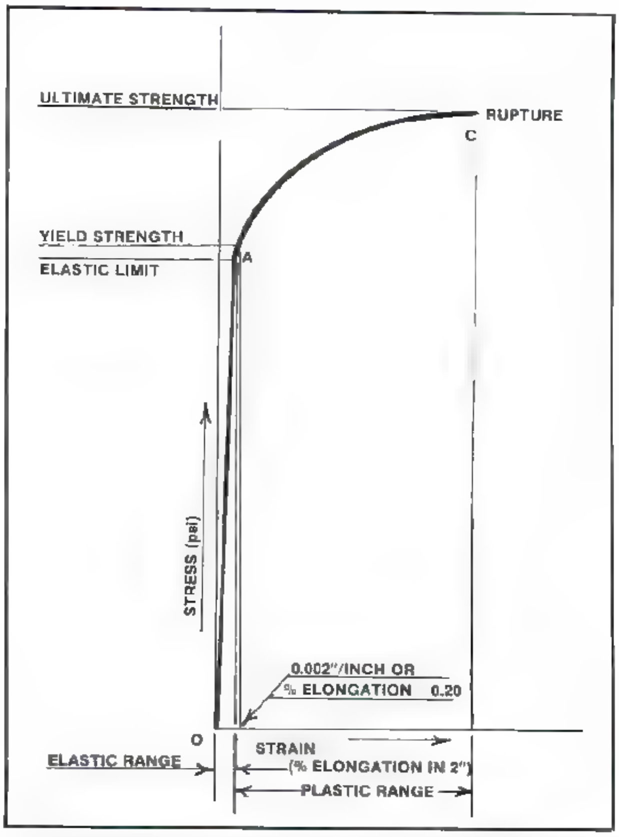
So, a load is an external force applied to a shape or a structure. An applied load will always create a stress within the material(s) of the shape/structure and that stress will always result in a strain or change in measurement of the shape.

Elasticity and plasticity

The behavior of elastic materials contrasts with that of plastic materials in that an elastic material that has been strained or distorted by the application of a load will return to its original size and shape when the stress produced by the load is removed. A plastic material will not. It will remain, at least to some extent, distorted when the load and the stress that it produced have been removed.



Load, tensile stress and strain.



Idealized graph of stress versus strain for a hypothetical metal.

Depending upon the level of stress imposed, many solid materials—especially metals—exhibit both plastic and elastic behavior. It is this duality in the nature of metals that allows us to use them as structural materials. If they weren't elastic they could not withstand the loads that we impose upon the structures that we build from them. Similarly, if they weren't plastic at some higher level of stress, we could not form them into satisfactory shapes from which to fabricate the structures.

Hooke's law

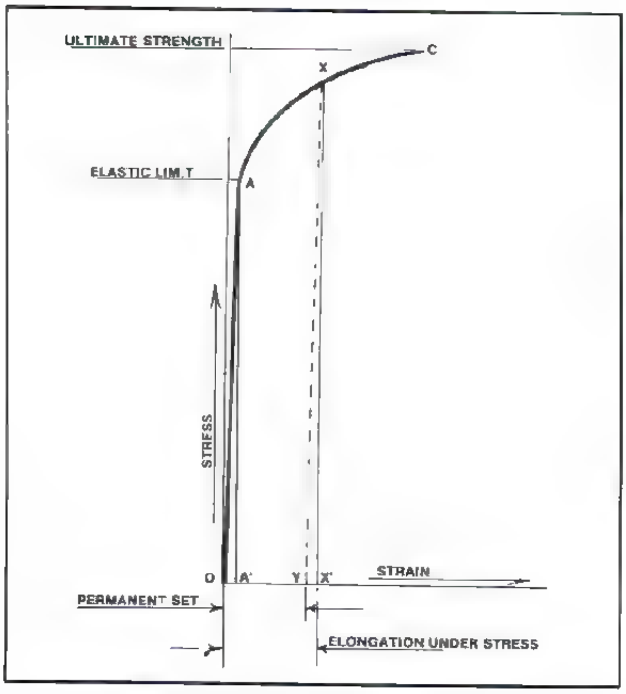
One of the first men to approach metallurgy from an analytical point of view was the English mathematician Thomas Hooke who, in 1680, formally stated what Engineering Man had intuitively known for millenia: "The strain of any material is proportional to the load applied to it." Today we have simplified Hooke's law to "the strain of any material must be proportional to the stress within it." In other words, the more you pull on something, the farther it will stretch or, the amount that a given object will deform under load is directly proportional to the size of the load.

If an applied tensile stress of x psi will stretch or compress a given test specimen one unit of length, then a stress of 1.5x will produce an elongation of 1.5 units, a stress of 2x will produce a deformation of two units and so on. Clever man, Mr. Hooke—with that one deceptively simple statement he formalized the basis of all structural engineering—bridges, ships, race cars, aircraft, space vehicles or whatever.

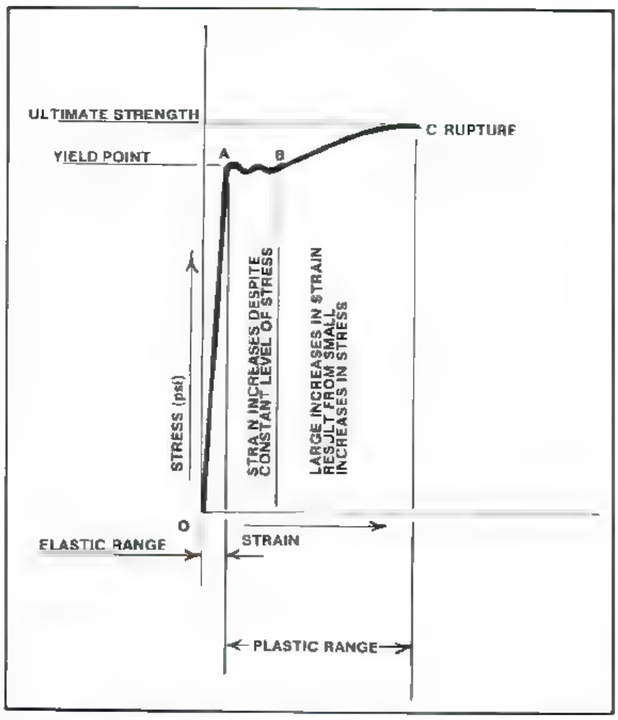
Hooke's law of proportionality, however, holds true only within those limits of both stress and strain which define the elastic range of any given solid material. The elastic range is that range of stress and strain within which the strained material (or a shape or a structure formed from the material) will return to its original size and shape when the load that produced the strain is released.

In the tensile testing of a material, a specimen bar of the material is mounted in a tensile testing machine and a steadily increasing tension load is applied until the specimen parts. While the level of stress is being increased, continuous and precise measurements of the specimen's length are taken. If we plot the actual relationship between stress and strain in a typical metal, we end up with a graph.

Note that the graph expresses stress, along the vertical axis, in psi, while strain, measured along the horizontal axis, is expressed in percent elongation. Percent elongation is the difference in length of a test specimen before it has been subjected to a tensile stress, and the length measured while it is subjected to a given level of tensile stress. Percent elongation is always expressed as a percentage of original specimen length (usually two inches). When we are speaking of the mechanical properties of a given metal or alloy, percent elongation is the difference in length between the original



Return curve of stress/strain diagram when load has been relaxed after stress has exceeded the elastic limit of the metal.



Idealized stress/strain diagram for mild steel showing the distinct yield point.

length and the length measured after the specimen has ruptured. It is used as a relative indication of ductility.

This graph is usually termed a stress/strain diagram and is characteristically made up of two distinct stages (labeled O-A and A-C on the graph). During the first stage, O-A, the curve is essentially a vertical straight line. Its slope is vastly exaggerated in the drawing simply so that the slope can be seen at all. For most metals the distance O-A along the horizontal axis of the graph is no more than one part in a thousand (0.001 in. per inch of specimen length, or a percent elongation of 0.1). The stress level at the corresponding point on the vertical axis is a function of the mechanical characteristics of the metal or alloy in question.

At any point within this area the material will always obey Hooke's law—if the applied load is released, the specimen will return to its precise original length and dimensions. This is termed the elastic stage of the material. Point A defines its upper limits of both stress and strain. In the stress department this point is termed the elastic limit of the material. It is defined as the maximum internal stress that a material can withstand without a permanent deformation remaining after the load has been released.

Accurate determination of the precise location of the elastic limit of a given material is difficult. However, it is a simple matter to measure the level of stress that will produce a specific elongation in any given material. For this reason, rather than attempting to measure the precise elastic limit of materials, metallurgists and engineers use the yield strength of a material (defined as the stress at which a material exhibits an arbitrarily chosen specified percent elongation: 0.002 in. per inch of original specimen length is the normally chosen figure) to indicate the upper limit of the elastic region.

The second stage of the stress/strain diagram, A-C, is called the plastic stage of the material. When the stress level falls within this area, the strained material will not return to its original dimensions upon release of the load but will exhibit some amount of permanent elongation or set. The curve ends abruptly at point C for the simple reason that, at this level of stress, the interatomic bonds holding the specimen together are stretched as far as they will go. The bonds run out of elasticity, snap, and the specimen ruptures in tension. The stress within the material at this point is termed the ultimate tensile strength of the material. It is defined as the maximum stress that the material can withstand without failure, and is expressed (in the United States) in psi.

If the load is released while the level of stress is between points A and C on the curve, the return graph to stable dimensions will be a straight line whose slope will be parallel to O-A, and the material will exhibit a permanent elongation or set even in the absence of stress. This is shown graphically as well. Note that the material, even after it has been strained beyond its elastic limit, still does its damnedest to return to its original dimensions. When stressed, the elongation is distance O-X' measured along the horizontal axis. When the stress is relaxed, the material contracts a distance of X'-Y. This results in a permanent set of distance O-Y.

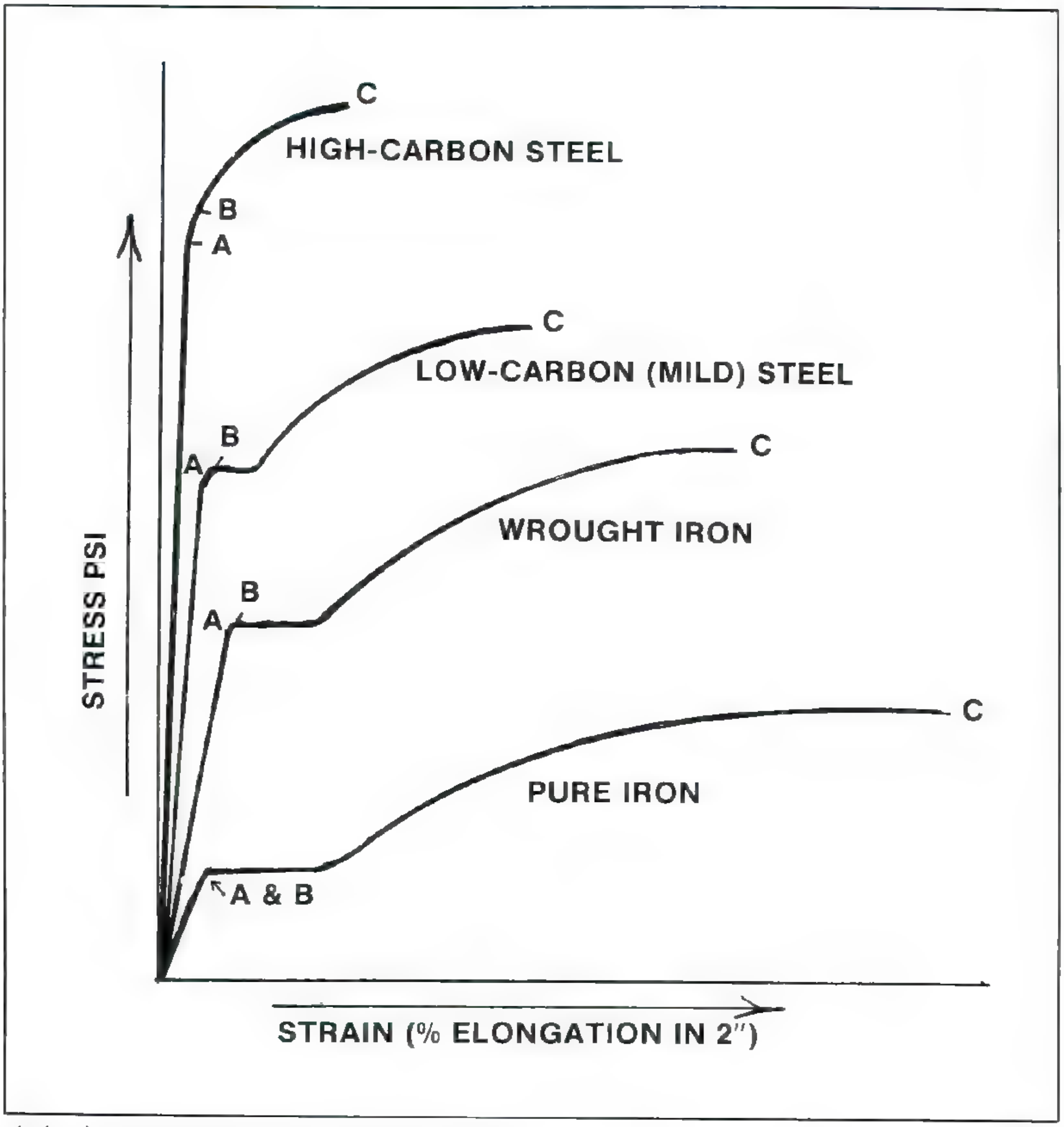
The yield point of a material is defined as that level of stress at which an increase in strain takes place without any increase in load. Some metals, most notably iron and its derivative, low-carbon mild steel, possess a sharply defined yield point. In this case, the stress/strain diagram exhibits a characteristic horizontal area (labeled A-B on the graph) in which things are somewhat confused in the stress and strain department. This area is essentially a horizontal line along which the metal will continue to elongate—even though the stress is held at a constant level. This characteristic of iron may have been sent to confuse us! None of the rest of our structural materials possess a distinct yield point and neither do the high-carbon and/or alloy steels. In fact, as shown here, the higher the carbon content of a given steel, the higher will be the ultimate tensile strength (more about this later) and the less distinct the yield point. At some level of carbon content, the distinct yield point will disappear altogether.

In the design and study of structures we are interested only in the behavior of our materials within the elastic stage represented by the area O-A in these illustrations. The plastic area is where metals are formed, not where they resist loads. If we should design a structure in such a way that the elastic limit of the material(s) chosen will be exceeded in service, then the structure will yield or fail under load and we will have either underestimated the loads involved or overestimated the strength of our materials. Either way we lose!

For small amounts of strain, the stretch/recover cycle is almost infinitely repeatable—the load and resultant stress can be applied and released literally millions of times with the same result. This elastic behavior of solids is the basis of all structural design and engineering. What is actually happening here is that the interatomic bonds that hold the atoms of all solid substances together are, to some extent, elastic. If we visualize the crystal space lattice of a typical metal as consisting of rigid atoms connected by coil springs which represent interatomic bonds, the whole thing will, I hope, become instantly clear—at least in principle.

From theory to practice: Mr. Hooke to Mr. Young

So far as I can determine, for 120 years Hooke's contribution to engineering had no practical impact at all. Finally Thomas Young, an English



Idealized stress/strain diagrams for various ferrous metals. In each case, A equals elastic limit, B equals yield strength and C equals ultimate tensile strength.

physicist, realized in 1800 that the elastic properties of a given material could be isolated from the behavior of a structure made from that material. He further discovered that each elastic material has its own unique constant of elasticity. This constant is a measure of the material's springiness, that is, the ability of the strained material to return to its original dimensions when the load is relaxed. The larger the modulus, the greater the stress that can be imposed without exceeding the elastic limits of the material. Young re-expressed and expanded Hooke's theory to state, "For any given material, stress divided by strain is equal to a constant."

So, Young's modulus (which is simply imposed stress divided by percent elongation) is a measure of the relative stiffness of a material. The stiffness of a shape or a structure is a function of both Young's modulus of the material(s) used and the shape of the structure. Young's modulus is often called the modulus of elasticity; the two terms are interchangeable. Constants are all very interesting, but to be useful they have to have some numerical value.

In order to arrive at a practical value for this dimensionless constant, the modulus of elasticity is defined as the stress required to double the original length of a test specimen of the material in question; that is, to achieve a strain of 100 percent. With any structural material this is a pretty damned big number—the modulus of elasticity of aluminum and its alloys is about 10,500,000 psi, while the modulus of elasticity for all steels is about 30,000,000 psi. Obviously any material strong and stiff enough to be used for structure will part or break long before its length is doubled under a tensile load. In fact, the elastic limit of all of our current structural materials is exceeded long before an elongation of even one percent is reached. After that, the material goes plastic and the structure goes limp.

Stiff does not imply strong

The modulus of elasticity of a material is, then, actually a measure of the stiffness or elasticity of the atomic bonds within that material and is therefore a measure of that material's relative stiffness. We do not build structures from materials with low moduli of elasticity simply because such structures would sag under any reasonable load.

The strength of a material, on the other hand, is a relative measure of the stress required to rupture the atomic bonds within the material. We do not make structures from weak materials simply because such a structure would break under load.

Together, the two properties of stiffness and strength define the physical properties of a solid material. For instance, carbon fiber is very strong and very stiff; steel is strong and stiff; copper is fairly strong and flexible; fiberglass made from chopped strand matt and polyester resin is weak and stiff; fiberglass made from unidirectional fibers and epoxy resin is strong and stiff; and gold is weak and flexible.

Summary

I think that a summary is in order here. A solid is considered to be elastic if, after a change of shape due to an external load, the body returns to its original size and shape when the load is relaxed. Plasticity, in the metallurgical sense, is the ability of a metal to be deformed beyond its range of elasticity without fracture; the result is a permanent change in shape. These two related properties are the most significant of all of the characteristics of the family of metals. Plasticity gives us the ability to form metals into useful shapes; elasticity allows us to use metal fabrications as load bearing members in our structures. Depending upon the level of stress imposed, all metals exhibit both elastic and plastic behavior.

In this book we are interested only in the properties of materials within their elastic range. This range is defined by the elastic limit of each material—that stress below which the material will obey Hooke's law of proportionality. Once again, Hooke discovered that the strain, or linear deformation, of any material must be proportional to the magnitude of the stress that causes it. Each metal—and each alloy—has its own characteristic modulus of elasticity which is defined as the stress that would be necessary to double the length of a test specimen of the material. Although this doubling is clearly impossible with any of the structural metals, the modulus serves as a useful indication of the relative stiffness of our structural materials.

While the stiffness of a material is clearly important to its suitability for use as a structural material, it is not the entire picture. We must also consider the strength of the material in question. But even that is not enough. Just because a material is stiff and strong does not necessarily mean that a structure made from it will be strong—at least in all circumstances. For instance, glass is very stiff and very strong but it is almost completely lacking in toughness. Consider, for example, a long, thin sheet of glass loaded as a beam and subjected to an unexpected impact load. The result would be shattering.

If you retain nothing more from this chapter other than a clear understanding of the concept that, in order to make an intelligent choice of materials for any structural purpose, you must always look at the whole picture, then my writing it and your reading it has been worthwhile.

2

Stress, strain, load and fatigue

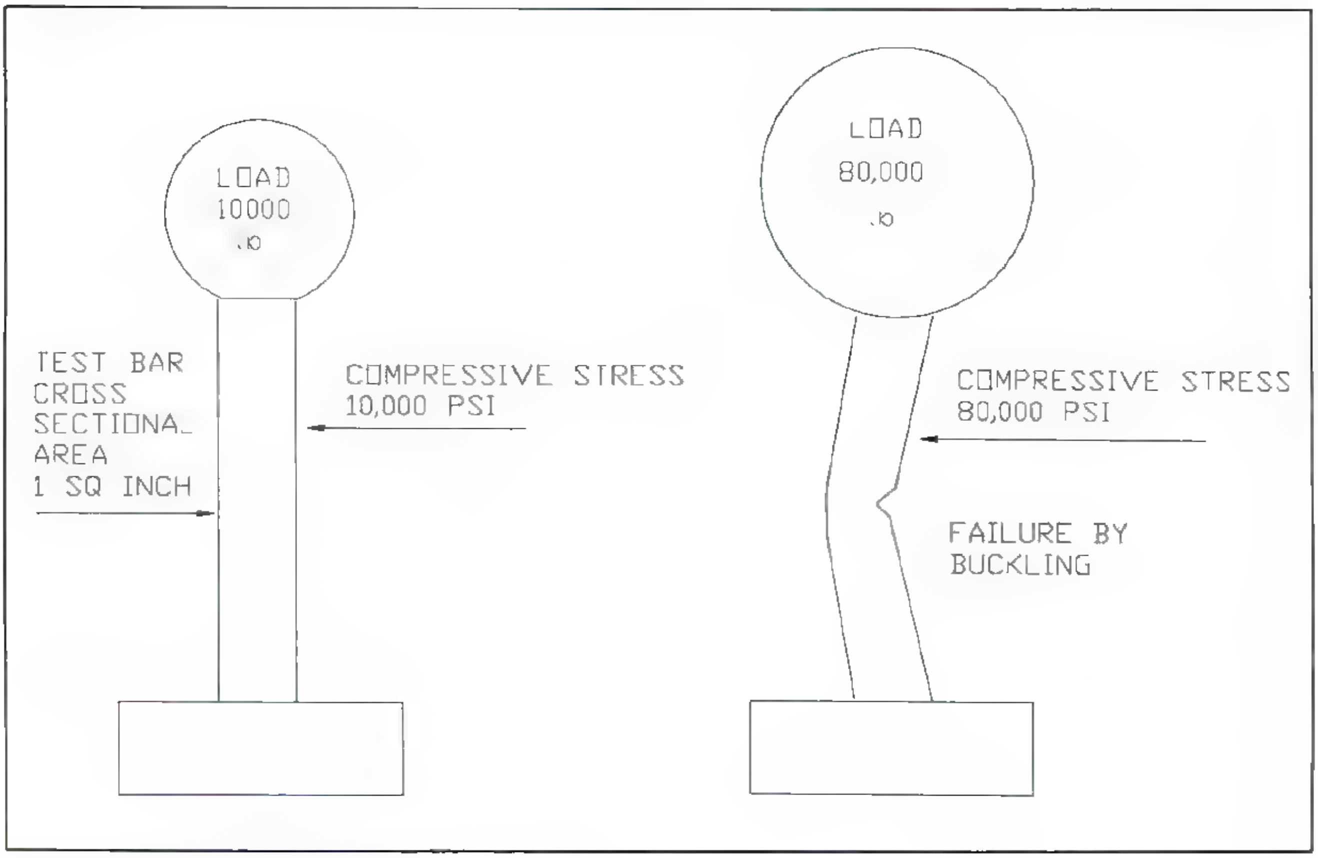
We have learned that, in engineer talk, load is the overall force to which a material or a structure is submitted in resisting an externally applied force. In the United States, load is usually expressed in pounds. We have also learned about stress. Stress is an internal force that tends to deform a body or structure; stress is the result of an applied load. It is usually expressed as unit stress—the applied load divided by the area over which it is applied. That area is the cross-sectional area of the material or the portion of the structure that is resisting the load. We express unit stress in psi. Strain is the elastic deformation of a body caused by stress. When the load is removed, the stress goes away and so does the strain.

Loads

In accordance with Smith's first law of simplicity, there are only four directions or paths in which we can feed a load into a piece of material or a structure. They are compression, tension, torsion and bending. This seems like an overly simplistic statement; it is not. Let's begin our look at the situation by using a solid, round steel bar with a cross-sectional area of one square inch and a load, W, as an example.

Compression loads

The illustration shows a simple compression load. The bar is positioned vertically so that it becomes a column, and the load is placed on top of it. The load pushes down along the axis of the bar

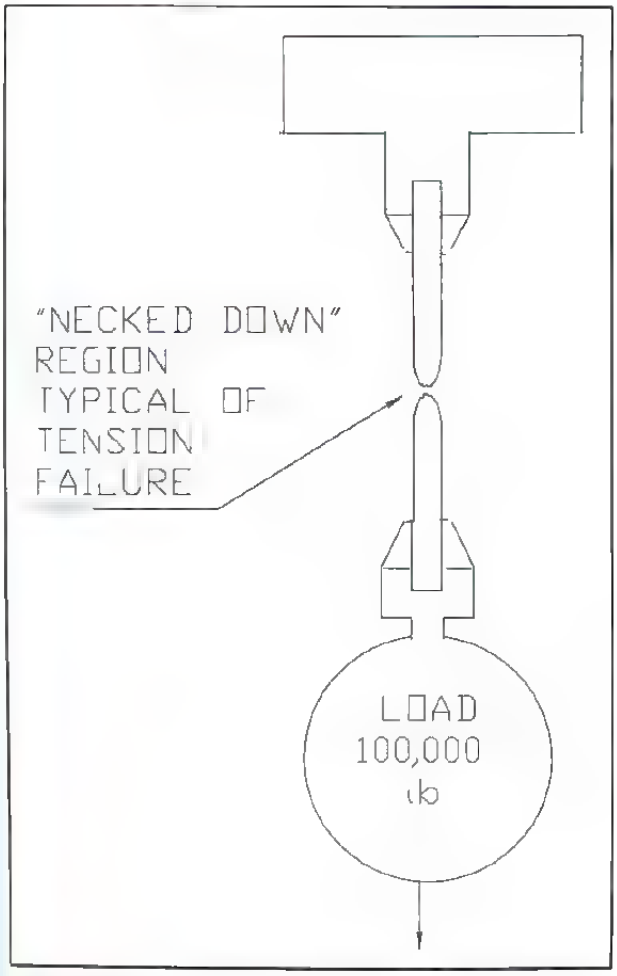


Compression load, stress and failure.

and attempts to shorten or compress it. If we increase the load until it exceeds the compressive strength of the bar, the bar will fail by buckling. In this book we will not be much interested in compression or in compression loads. This is not because our parts do not undergo compression stresses (think about a connecting rod or a trailing arm) but because, usually, if we design the part so that it is strong enough in tension, it will almost automatically be strong enough in compression.

Tension loads

The diagram illustrates a tension load. The same bar is now suspended from a support and the same load is applied—this time at the bottom of the bar. In this case the load (still acting along the axis of the member) tries to stretch the bar. When the load has been increased sufficiently, failure will be by elongation beyond the elastic limit of the material and eventual rupture. With elastic materials, tensile failure will always by preceded by a reduction in cross-sectional area—called necking down—and a permanent (or plastic) stretching or elongation of the material. As long as we are dealing with structures or mechanisms, we will be very much concerned with tension loads.

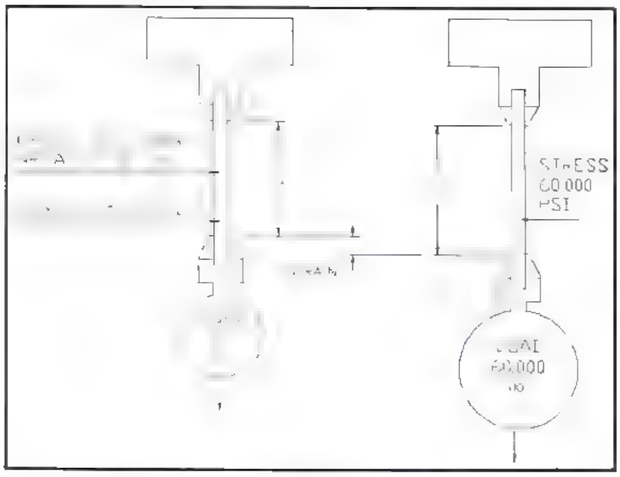


Tension failure.

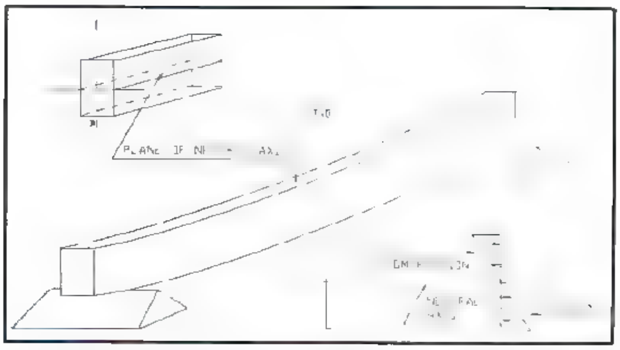
Bending loads

An example of a simple bending load would be a beam supported at each end and loaded in the middle. When we apply the load, the beam, because it is elastic, bends into a curve. But the action is not as simple as it may seem. The upper surface becomes concave, and the fibers of that portion of the beam are stressed in compression. The lower surface, on the other hand, stretches in reaction to the load and these stretched fibers are stressed in tension. Somewhere in the cross section of the beam there will be a transitional plane where there is neither compressive nor tensile stress. This plane is termed the neutral axis of the body. In the case of regular shapes—square, rectangular, round or whatever, either solid or hollow, so long as the wall thickness is constant—the neutral axis will run through the geometric center of the shape.

In the case of complex shapes, the precise location of the neutral axis is more difficult to determine, but it will be there. In any case, as we move away from the neutral axis, compressive stress will



Tension load and stress.



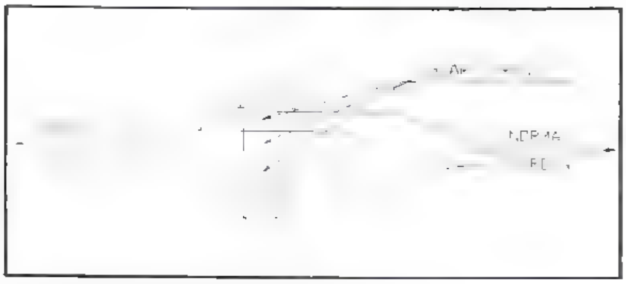
The neutral axis and the distribution of stress within a beam loaded in bending.

increase as we move toward the load and tensile stress, and its resulting strain will increase as we move away from the load. No matter what type of bending load we are concerned with, when the load is increased sufficiently the initial failure will occur by buckling of those fibers that are in compression.

Normal and shear stress

In engineer talk, a force acting at right angles to an axis is termed a normal force. The stresses produced in a material or in a structural member by either a tension or a compression load will necessarily be perpendicular to the cross section of the member. Compression and tension stresses are therefore termed normal stresses.

There is, however, no law that says that loads have to be applied along the axis of the member. The load is often applied at some angle to the axis. For example, when two bars loaded in tension are connected through a fork and eye, the tensile force is transmitted from one bar to the other from the two legs of the fork end of the first bar to the eye of the other through the connecting bolt. In this case the load will produce a stress across or through the cross section of the bolt. This stress in the plane of the cross section is termed a shear stress.

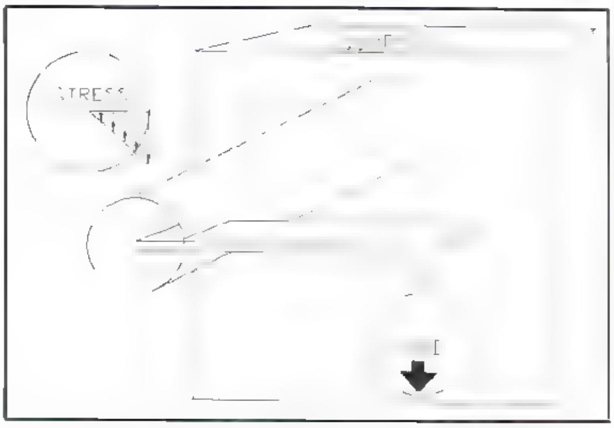


Normal stresses and sheer stress.

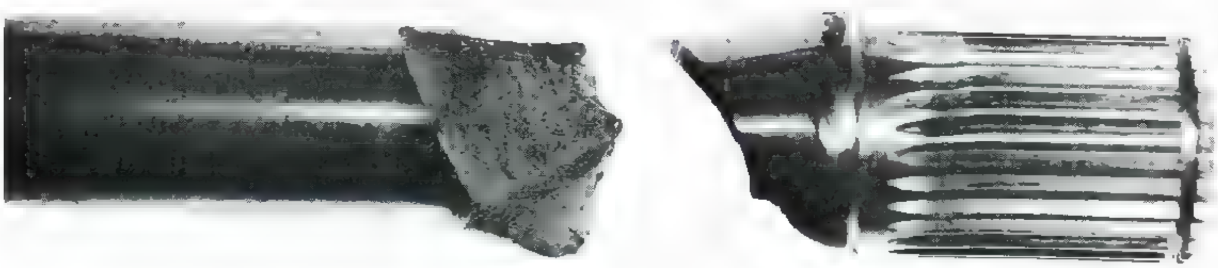
Torsional loads

To load the bar in torsion, we clamp one end, and apply a twisting force to the other end. The load will be resisted by shear stress in the plane of the cross section of the member. If we look at the cross section and visualize what is happening as the bar twists, we will see that not very much at all happens toward the center of the bar. In fact, at the exact center of the bar there is zero stress (and zero strain). The center of a bar loaded in torsion is, in fact, the neutral axis of the bar. As you move out from the center, both the shear stress and the resulting strain increase and reach some maximum value at the outside surface of the bar. This is, of course, why all of our torsional anti-roll bars and suspension torsion bars are tubular in form.

If we load the bar to failure, it will rupture along a spiral path at a 45 degree angle to the axis of the part. This characteristic spiral fracture path is typical of the torsional failure due to shear stresses.



A shaft loaded in torsion and the distribution of shear stress within the shaft.



The spiral fracture path of failure due to shear stress from torsional load.

That's it! That is all there is to understanding load. You can combine either tension or compressive loads with bending and torsional loads, and you can apply them from different directions or along various axes, but you cannot come up with any more types of loads. The stress story is even more limited—although it is nowhere near as simple.

Stress

We have seen that when a beam is loaded in bending there is compressive stress in one portion of the beam and there is tensile stress in another part of the same beam under the same condition of load. We have also learned that the two portions of the beam are separated by a plane where there is no stress. The natural question at this point is, What about the bending stresses? Further, if, when we twist a bar, the shear stress in the material increases from the center outboard, where are the torsional stresses? The answer is deceptively simple—there is no such thing as bending stress, or torsional stress. All stresses are either normal or shear in nature. All structural loads are converted (by Mother Nature, not by us) into either normal—compressive or tensile—or into shear stress.

There is, however, one last type of stress that I should mention. Residual stresses are those stresses that are present within the crystal structure of an unloaded material that has been strained by cold or hot working—or by welding. Residual stresses are independent of and cumulative with working stresses. Unless we are aware of their existence, they can provide some nasty surprises. Undesired residual stresses can usually be relieved by annealing or by normalizing.

Later in this book we will see that not all residual stresses are undesirable. In fact, the operating principle of the threaded fastener depends upon the premeditated creation and use of residual stress.

Again, the basics really are that simple. I should probably point out that the actual stresses in structural members are unlikely to be simple. In the real world, stresses are usually a complex combination of shear and normal stresses acting along multiple axes.

Metal fatigue—Or why parts break

Most of the metal parts that break do not fail from being overloaded. They fail from metal fatigue. This includes hardware. Most people seem to have no clear idea of what metal fatigue is or how it works. If we are going to avoid it, we had better learn to recognize it—so here goes.

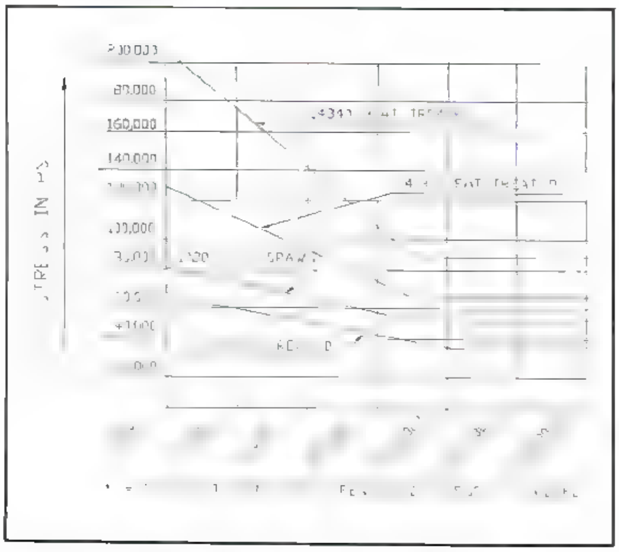
History

By the middle of the eighteenth century, engineers had learned to accurately estimate the static strengths of the materials that were available to them, and to use those materials in the design and construction of structures that would usually withstand the loads that the designers expected would be imposed upon them. The materials varied from stone and the hard and soft woods (bamboo in the Orient—an amazing material), to bronze, cast iron, wrought iron and small quantities of precious carbon steel. Failure of well-designed structures was rare (if spectacular), and some of the structures were pretty damned impressive—aqueducts, bridges, cathedrals and sailing ships come to mind. Given the rate at which our freeway overpasses and bridges are collapsing, I am not sure that we are doing significantly better today.

James Watt, the brilliant eighteenth century Scot engineer (not our former Secretary of the Interior), changed all of that! His development of the practical condensing steam engine in 1769 changed the nature of our world and the structure of our society forever—the world of engineering most of all. It also began more than two centuries of spectacular structural failures!

By 1781 Watt had devised mechanisms to convert the reciprocating action of the steam piston to the rotary and linear motions required by industry. The age of mechanization had begun. It was this mechanization, not the industrial revolution that spawned it, that gave birth to the mixed benefits of our present industrialized society—and to the twin sciences of engineering and physical metallurgy as we know them today.

With the advent of steam power, various pieces of machinery began to move at reasonable speeds and under reasonable pressures, as opposed to the dignified but dead slow pace of water and wind power. The traditional design parameters for structures and mechanisms, which had been based solely upon the ultimate strengths of the materials involved, quickly proved to be inadequate. The same was true of many of the materials themselves.



Fatigue limit curves for some common steels.

A period of some confusion—not to mention excitement and physical danger—followed. Eventually the engineers began to realize that, while a structure subjected to a relatively constant load may last virtually forever, any member that is subjected to significant variations in load is liable to fail unexpectedly even though it has never experienced a load that even approached its static capacity.

To state the situation more simply, under repeated cyclic (as opposed to continuous) stress the capacity of a metal to withstand stress gradually diminishes and, in most cases, cannot be restored. Metals that are subjected to fluctuating loads can and do break after a finite number of load cycles (or more accurately, stress cycles) during which the loads applied and the resultant stresses imposed remained below the ultimate strength of the metal. This type of failure is termed fatigue failure. For the past two centuries the nature, prediction, detection and prevention of failure due to metal fatigue has been the subject of a great deal of study—most of it beyond the scope of this book. In the simple interest of survival, however, each of us must understand the basics of metal fatigue.

Fatigue is estimated to be the primary culprit in more than ninety percent of all in-service failures of metal parts. This is why we express the expected life of critical aircraft and racing car components in terms of hours of service. Realistically we should express the life span in terms of stress levels and load cycles, but service hours, being easy to measure and keep track of, are a lot more practical.

The aerospace industry devotes a lot of time, energy and money to the computation (i.e., estimation) of expected average stress and cycles per service hour. They also install recording devices and overload indicators. As a result, their record in this respect is very good indeed—although when they do have a failure it is liable to be a beauty. For the most part, the rest of us guess at all of this and do a lot of inspecting. Perhaps surprisingly, our record is also pretty good.

I should point out that, in the world that I live in, there is no practical way to predict, or even to measure, the stresses involved in or the frequency of occurrence of operator induced overloads—hard landings, curb cloutings, over-revs and the like. The more competent and experienced estimators (i.e., engineers) involved are only too well aware of this and tend to apply safety factors of dubious accuracy and generous proportions. Actually, what we engineers call the factor of safety in our calculations could, with some accuracy, be termed the factor of ignorance.

In this age of the microprocessor, the crystal ball is still a valid and indispensible engineering tool. Knowing this, the more intelligent operators do their best to prevent the crystal ball from becoming cloudy by informing us estimators when they have hit things or otherwise overstressed the machinery. This is one of the major differences between the professional racing driver and the would-be professional.

The nature of fatigue failure

So, what actually happens when a metal part fails from fatigue? *Nothing very sudden!* This may sound glib, but it is, in fact, crucial to understanding the nature of fatigue and to the avoidance of fatigue-induced failure. With a stud failure due to fatigue, the process begins with a tiny crack at the point labeled A on the surface of the shaft in the photo. Assuming a sound metallurgical structure to begin with, the initial fault will always be, thank God, on the outer surface of the part—simply because that is where the point of maximum stress must be located.

The initial crack is started by repeated application of an imposed stress. The crack can be stopped (or at least its progress temporarily halted) by two factors. First, since the load is cyclic in nature, when the load decreases, the stress will be reduced to a level that the material can withstand. If it is not reduced to this level the part will fail immediately and fatigue will not be a problem. Second, the presence of the crack disturbs the natural orderliness of the metal's crystal space lattice. This causes the natural dislocations within the lattice to migrate to the leading edge of the crack and to block the further progress of the crack through the material—like ants rushing to repair damage to their nest.

This is fine for the moment, but there is a long-term problem. The initial crack has reduced the cross-sectional area of the member. Consequently there is less material available to resist the next load cycle. We know that unit stress is load divided by cross-sectional area. Therefore, the next time the load is applied, the imposed unit stress will be greater than it was the last time (assuming an equal load). More important, despite the migrated dislocations, the jagged bottom of the initial crack acts as a first-class stress concentrator. Just as the wolves always go after the weakest member of the herd, as more load cycles are applied, the concentration of stress at the leading edge of the crack will cause the crack to enlarge until enough fresh material is engaged to resist the stress—this time. These fatigue cracks are transgranular in nature; the fracture actually splits the individual grains of the metal rather than following grain boundaries. As a result, the opposing surfaces of the crack tend to be quite smooth in appearance.

As the load (and the stress) continues to cycle, the crack will progress in a crack/pause/crack/pause sequence. The crack will remain stable (perhaps for many hundreds, or thousands, or even millions of cycles) until the stress level reaches the point where the remaining material is no longer able to resist the load. The crack then restarts and progresses until the load is reduced. The disloca-

tions then do their thing, and enough fresh metal is again exposed to temporarily resist the load. The crack stabilizes and the cycle begins again.

This repeated sequence of events creates the typical smooth opposing surfaces with telltale concentric beach marks that are also visible in the illustration. The beach marks are formed by the progressive enlargement of the crack and radiate outward from the focus of the original fault. Beach marks are characteristic of the fatigue failure. Eventually the crack progresses through the material to the point that the remaining metal is no longer able to withstand the stress imposed. The next sufficiently large load application produces sudden and catastrophic failure of the remaining portion of the metal. This last failure is intergranular in nature.

The final rupture proceeds *along* the grain boundaries, leaving the individual crystals exposed as a rough and granular surface. This allows the self-appointed expert to peer knowingly at the part and proclaim, "Ha! Just as I suspected—a crystallization failure." When you hear this sort of statement, strike the proclaiming person from your list of those to be consulted in the future. He (or she) is as phony as the proverbial three dollar bill. Since all metals are composed of crystals, all metal failures are crystalline in nature. It is merely a question of whether the grains were progressively split by fatigue or torn apart by catastrophic failure.

Detection and prevention of metal fatigue

By now, three important facts have become evident. First, since the fatigue failure of any metal part must initiate on the outside surface of the part, if we inspect our parts often enough (and well enough) the odds of detecting an incipient fatigue failure before it causes a catastrophe are very good—if we know where to look and what to look for. This is what dye penetrant, magnetic particle, sonic and X-ray inspection are all about. It is also what cleanliness, the human eyeball and old-fashioned common sense are all about.

The second fact is that if we can somehow inhibit the formation of cracks on the surfaces of our parts, the fatigue life of the parts will be greatly extended. This is why we polish parts and go to great lengths to avoid surface scratches. It is also why we shot peen critical parts.

Third, just because a part passes inspection today does not mean that it will pass tomorrow. There is a finite point in the fatigue life of every metal part when the first fatigue crack starts. Fortunately, with the exception of very highly stressed parts, or those that undergo cycling at extremely high frequencies (as in valve springs), we will normally have time to detect the fault before failure occurs—if you inspect often enough and well enough.

It is important to realize that, in the fatigue history of any part, neither the rate at which stress is built up within the part nor the period of time over which the stress is maintained is significant. Only the levels of stress experienced, the type of stress(es) and the cumulative number of stress cycles are significant. The higher the stress, the fewer will be the number of cycles required to produce failure, and vice versa.

If you intend to bend a wire coat hanger back and forth until it breaks, the farther you bend it

Stud that has failed from fatigue, showing "beachhead" marks of progressive cracking and the rough surface of final catastrophic failure. Roy Kiesling

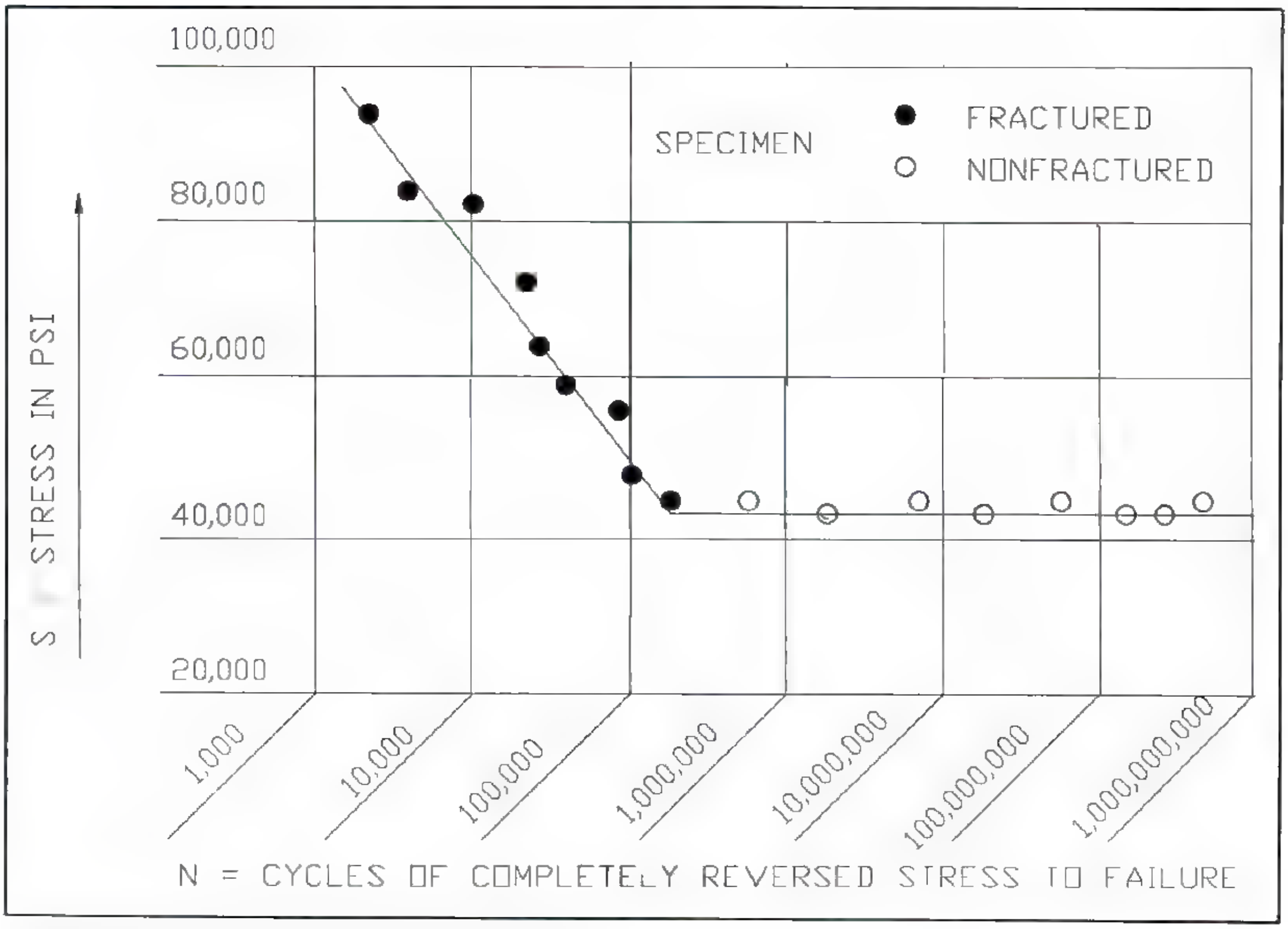
Driveshaft that failed from sudden, one-time catastrophic over load. Roy Kiesling

each time, the fewer number of times you will have to bend it before it breaks. The speed with which you bend it (so long as it is low enough not to produce significant heat) doesn't matter. Neither does the amount of time that you rest between efforts. If you bend it 90 degrees once a week, it will fail at exactly the same number of cycles as if you had bent it 90 degrees once a minute. Bending it through 45 degrees instead of 90 will more than double the number of cycles to failure, and bending it through 180 degrees instead of 90 will reduce its cyclic life by more than half. The fatigue characteristics (as well as the hardness and strength) of all metals deteriorate with heat—sometimes drastically, depending on the alloy involved and its heat treatment.

Fatigue limit—How long will it last?

When a graph of cycles to failure at various stress levels is plotted—as was done for a bar of E 4130-N chrome-moly steel in the graph that appears here—it will show that at some level of stress the test specimens will stop breaking, no matter how many cycles they are subjected to. At this point, the curve becomes horizontal, or nearly so. The fatigue limit of a material is indicated by the point where the curve approaches the horizontal. It is defined as the maximum stress at which the material will withstand an infinite number of completely reversed cycles of stress. It is normally assumed that if a specimen can survive several million stress cycles, it can survive forever.

There is no direct relationship between the ultimate strength, the yield strength and the fatigue limit of materials. We can, however, generalize and say that, for most alloy steels, the fatigue limit in reversed tension/compression is usually about fifty percent of the ultimate tension strength (UTS). Because a torsional load produces a shear stress and because our materials are nowhere near as strong in shear as they are in tension or compression, the fatigue limit for steels in reversed torsion is only about fifty percent of that for the same steels in reversed tension/compression—or approximately twenty-five percent of the UTS. In the case of parts made from alloy steels, the first detectable fatigue crack will show up at about eighty to ninety percent of the fatigue life of the part. Assuming a reason-



The fatigue limit curve of SAE 4130 chrome-moly steel.

able mechanical design, this means that frequent and conscientious inspection will almost always find the flaw before failure occurs.

Beware of experimental data—
Or never trust the experts

Test figures—all of them—ultimate and yield tensile strengths, shear strengths, fatigue limits and endurance limits look really good in the tables. The tables are in books. Books are made from paper. Paper does a truly poor job of cushioning the impact between metal (or flesh) and hard objects such as stone or concrete.

The figures are derived in laboratories from smooth and perfect test specimens under precisely controlled conditions of load and environment. The test specimens are necessarily small in diameter. While the exact nature of the effect of physical size on the fatigue life of metals is somewhat confused and unclear, it is damned certain that the larger sections actually used in our structures have a notably shorter fatigue life than the laboratory specimens from which the tables are constructed.

We live in the real world. Our machining, our fabrication, our heat treating and our joining are imperfect at best. Our parts inevitably wind up with scratches that serve as stress raisers and we make mistakes in design, execution and application. Sometimes we make mistakes in all three on the same part! Our fasteners are subject to loosening in service, and all of our structural members are exposed to corrosion. Further, while the prediction of the fatigue behavior of metals as test specimens is really quite simple and reasonably accurate, the same cannot be said of the prediction of the fatigue behavior of structures—especially of complex structures. Lastly, Murphy guarantees that we will eventually overload every part that we make.

It should now be obvious that stress analysis and fatigue prediction is a pretty complicated subject and that it is well beyond the scope of this book. Very true!

What you need to know

Most of us have no real need to know either the details of stress analysis or the calculation of fatigue strength. What we do need to know is what to watch out for. For example, we have to watch out for (and avoid like the plague) concentrations of stress. Basically, a stress concentration will occur at any point where the free flow of stress along the component is interrupted or piled up by an obstacle—such as a hole, a notch, a scratch or a change of section. The obstacles are commonly called stress raisers and are to be avoided whenever possible and minimized when avoidance is not possible. Both avoidance and minimization require recognition. Fortunately, recognition requires little more than common sense.

The flow of stress around a stress raiser in a metal component under load can be likened to the flow of water around a rock in a stream: the flow will always be smoother if the rock is round. We will divide stress raisers into two basic types: the obvious, and the not so obvious.

Obvious stress raisers

Notch

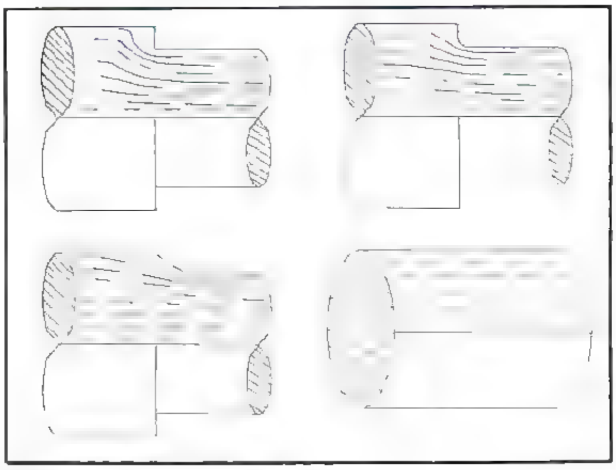
The term notch can be applied to all kinds of irregularities in structure—bumps, grooves, scratches and such. Notches can be found in machined parts, fabricated parts, weldments, castings and forgings as well as on drawings. Some cannot be avoided and are carefully placed and configured by knowledgeable designers. Some are put there on purpose by people who are ignorant. Others are created accidentally by people who are merely careless.

The most common type of notch is the scribe mark. Scribing any line on metal, except a line to be completely cut away, is a cardinal sin and an absolute guarantee of early failure. The failure will take place right along the scribed line, which might just as well be marked, tear here. Self-trained metal fabricators are particularly adept at this one. A poor surface finish on a lathe-turned part (or, for that matter on any machined part) counts as a series of notches, as do any tool marks or scratches on the surface of a part. Any imperfection in the root of a thread constitutes a super-notch.

Unfilled hole

The drilled lightening hole so loved by hot rodders and low-level racers is little more than a death trap of stress concentration. The drawing here shows what happens to the stress curve of a bar when a hole is drilled in its center. Even though the hole is deburred, the result is disastrous. In the case of riveted structure it is essential that rivet holes are precisely sized and aligned between sheets, and that the rivets fill the holes.

As a point of interest, the formation of fatigue cracks at the edges of drilled holes can be signifi-



The flow of stress concept.

cantly delayed by the simple act of peening the edges of the hole. The edge of the hole is deburred by chamfering, and a ball bearing of suitable size is placed on the chamfer and walloped with a hammer. It is that simple. The peening produces a compressed surface layer which inhibits the formation of surface cracks. It also creates a raised lip at the outside edge of the chamfer which must be filed smooth.

Not-so-obvious stress raisers

Joints

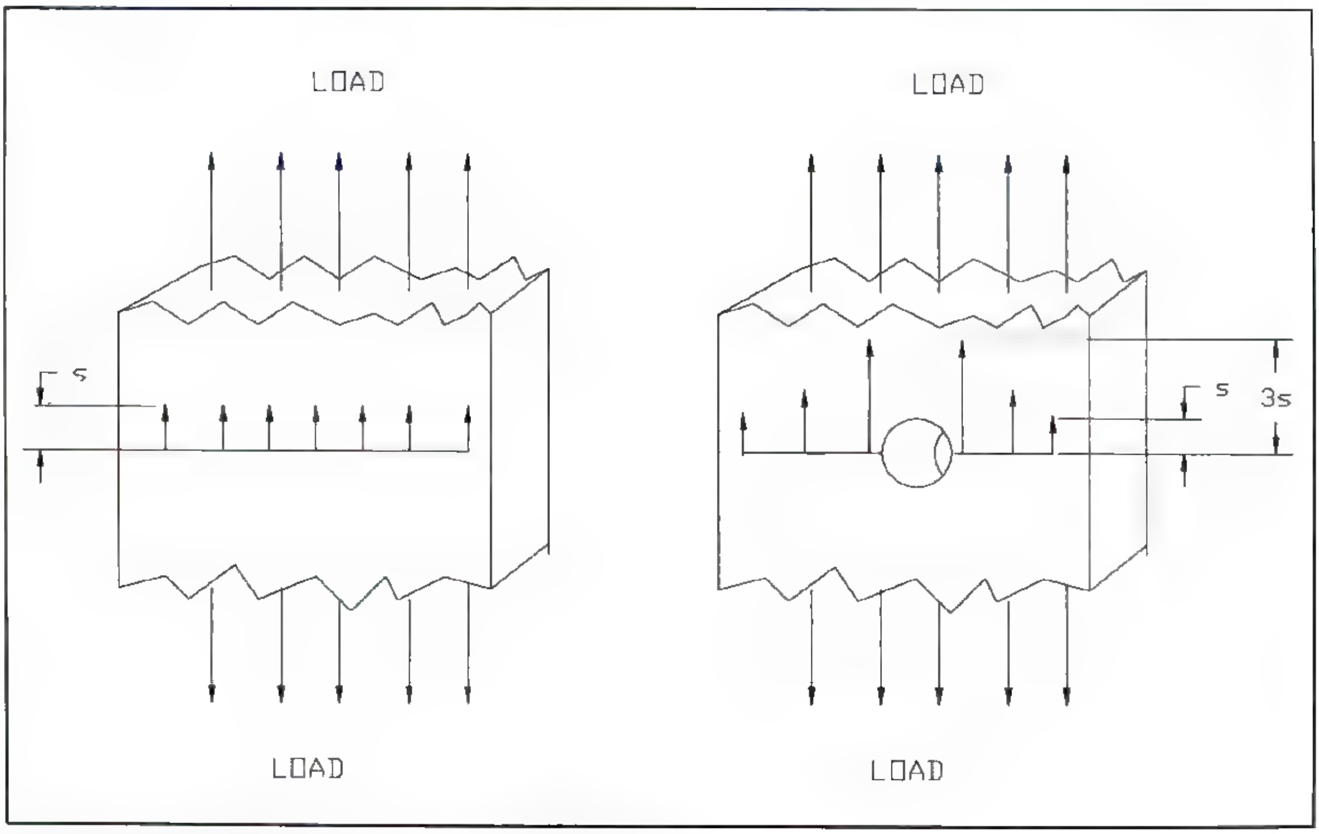
Any type of joint forms a barrier to the smooth flow of stress. Therefore, from both the strength and the fatigue points of view, joints in any structure are bad. They are also a necessity. The trick is to design our joints with the flow of stress and fatigue in mind. We will cover joint design later.

Hardness

In general, the stronger and harder a steel, the higher its fatigue limit will be *but* the more notch sensitive it will be. Stress raisers that would be trivial in SAE 1018 low-carbon steel become critical in high-strength 4340 M. In fact, not only are the harder steels more prone to the formation of the initial fatigue crack than the softer ones but, once that first tiny crack has formed, it will propagate much more quickly in a hard steel than it would in a softer and more ductile steel. So the use of a super steel is no substitute for either good design or good manufacturing practice.

If we are not going to take extreme pains to avoid the creation of even minute stress raisers in design and manufacture to obtain a good surface finish, and to protect the part from corrosion and surface nicks in service, then there is no sense at all in selecting a high-alloy steel or a high-heat treat. In fact, unless we do all of that, the fatigue limit of a part made from the good stuff may actually be lower than if we had made it from a carbon or a low-alloy steel. Unless great care is exercised, once we have hardened a steel to an ultimate tensile strength of about 150,000 psi, the increase in strength and fatigue limit that results from further hardening is often offset by the increase in notch sensitivity. It is worth using the higher-alloy steels, however, and taking the care required.

These are some of the problems you are liable to encounter with the super-whatever, "superior to SAE Grade 8" bolts that are on the market. The only way in which they are superior to SAE Grade 8 is that they are harder. Since they are harder, they are more brittle. They are also liable to be full of stress raisers. More on this later.



The distribution of tension stress in a steel bar with a central round deburred hole.

Surface finish

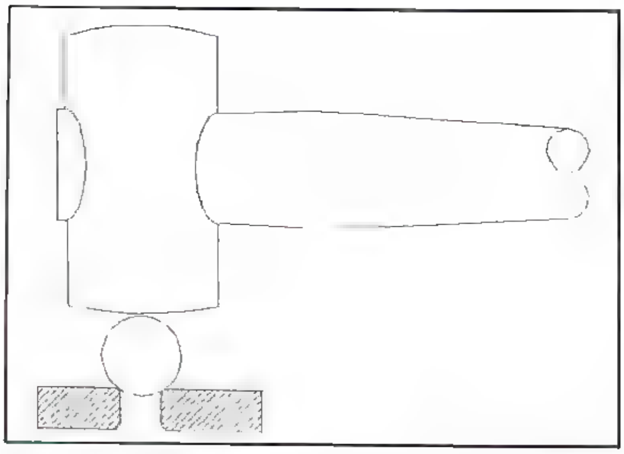
What lies beneath the surface of the metal largely determines the ultimate strength of the material, its toughness, its impact resistance and even the rate at which cracks will propagate. It has little to do with the initiation of the fatigue failure, however. Given a sound metallurgical structure, the first crack will always be on the surface. It will appear at that point on the surface with the highest concentration of stress. One obvious way to extend the fatigue life of a highly stressed part, then, is to make sure that the part has a smooth surface finish all over. The chart illustrates the effect of various surface finishes on the fatigue life of a test specimen.

Change of section

Anytime there is a section change, no matter how small, a fillet radius is required—and the radius must blend smoothly into both adjacent surfaces. There can be no exceptions if we expect the part to live under conditions of repeated stress. This is particularly true for the roots of threads and for the intersection between the heads of bolts and their shanks.

Corrosion

Corrosion is the alteration of the structure of metals by chemical attack. It always begins at the surface of the metal. Our structural metals, and their alloys, exist only in a forced or artificial state of stability. In nature they are found only as oxides (ores) and given any opportunity at all, they will return to that state. The actual corrosion process is complex, but it begins with the loss of ions from the crystal lattice. The ions are always lost from positions of maximum stress. This loss of ions will lead to weakening and eventual destruction of the lattice, if not halted, as is evident to anyone who has ever seen a rusted ferrous part, like an old tin can in the woods, or the bodywork of a street car that has been through a couple of eastern winters.



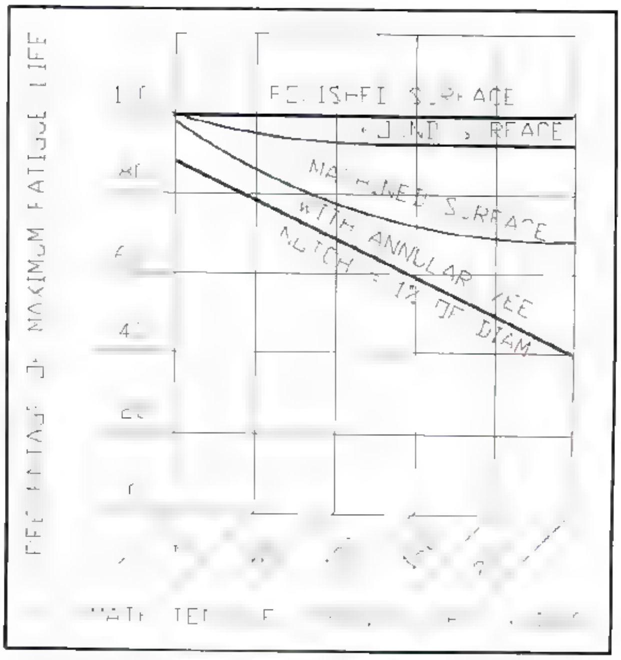
Peening a drilled and chamfered hole to form a compressive layer and delay the onset of cracking.

What is not so evident is that the initial corrosion pits in the surface of any metal serve as remarkably efficient stress raisers, for two reasons: First, they form at the position of maximum stress on the surface of the metal—in other words, where the metal is most vulnerable to the formation of fatigue cracks; and second, they are jagged as hell.

Avoidance of stress raisers

In many branches of engineering, fatigue failure can be irritating, embarrassing and even costly—in terms of downtime for replacement of failed parts. With racing cars and with aircraft, we are not talking about embarrassing or expensive, we are talking about deadly. For this reason, if for no other, we should try to exercise standards and practices equal to those developed (and, hopefully, practiced) by the aerospace industry. Since aerospace has spent a great deal of time and money amassing a vast amount of experience in the field, it makes sense to borrow as much of their knowledge and expertise as we can. They are very good at avoiding failure due to metal fatigue.

The history of fatigue failure in the aircraft industry is a history of nicks, notches, gouges, sharp corners, section changes, keyways, oiling holes, inclusions (including welding slags), corrosion pits and other such crimes against nature. It is, in short, a history of human ignorance, laziness and carelessness. It is most emphatically not a history of inadequate metals, metallurgy or even of inadequate application. More than ninety-nine percent of the times that metal has failed in service from



The effect of surface finish on the fatigue life of laboratory test specimens.

fatigue, it has done so because human error made failure inevitable.

Trivia

In the avoidance of metal fatigue there is no such thing as a trivial detail. One of the best illustrations of this fact that I know of involves the fatal crash of a military aircraft in the 1950s. This particular crash was caused by the fatigue failure of a propeller blade. The fatigue failure was initiated by the ill-chosen location of a stamped serial number.

Summary

Any metal component subjected to cyclic stress can fail from fatigue even though it has never been subjected to a load sufficient to impose a stress near the ultimate strength of the material. Fatigue failure will always begin with a tiny crack originating at the one spot where, for whatever reason, the metal is repeatedly stressed beyond its local endurance level. This spot will almost always be on the outside surface of the part. It will always be a stress raiser of one sort or another (even if it is only the least perfect spot on a well-finished part).

The fatigue crack is always gradual and progressive in nature. This means that it can almost always be detected—prior to failure of the part—by a trained and curious eye with a bit of help from nondestructive testing equipment. Lastly, the formation of the fatigue crack could almost certainly have been prevented by one or more of the following persons: the designer, the machinist, the fabricator, the welder or the mechanic.

It follows then, that the most effective way to avoid fatigue failure of metal components subjected to cyclic stress is to avoid the creation of stress raisers. The places to avoid their creation are: First, on the drawing board, since it is always easier (and cheaper) to design a part correctly than it is to redesign and remanufacture it; second, in the shop, since it is always easier to make it right the first time: "We haven't got time to make it right, but we've got time to make it twice!" is all too frequently true; and third, during inspection—either before the part is assembled/installed or during routine maintenance.

Why bother looking?

The first step in the prevention of the reoccurrence of a component failure is the identification of its cause. If we look closely enough at any part that has failed from fatigue (assuming that the part has not been peened or beaten beyond legibility by the flailing that followed the failure), we will almost always find the characteristic beach marks produced by the progressive enlargement of the crack. If the beach marks themselves have been obliterated by the polishing action of the opposing surfaces of the crack, you will find the telltale smooth surfaces where they were. If the part was designed and built with a large margin of safety, the smoothly radiating beach marks may comprise a notable percentage of the total cross-sectional area of the part. When a part is designed and built with a low margin of safety, the beach marks may be few and may not take up much of a percentage of the cross-sectional area of the part. But if the failure was initiated by fatigue, they will be there.

The subject of stress raisers is something like the subject of giving advice to a son: It's easy to become tedious and it is simply not possible to describe every potential pitfall. An instructor must depend on a combination of the student's intelligence and his remembrance of specific examples to keep the pupil out of trouble until he has developed an eye for stress concentrations.

3

Thread physics

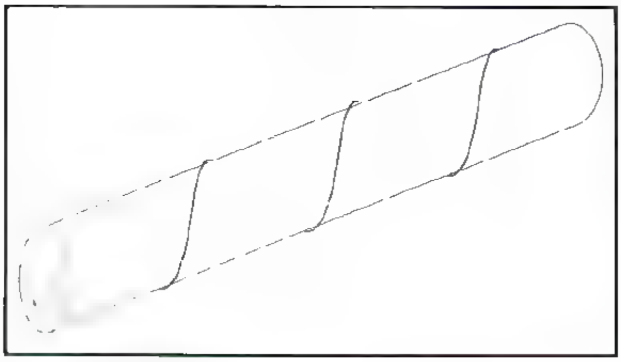
The joining of materials

A fastener can be defined as a mechanical device for holding two or more bodies in definite positions with relation to each other. Fasteners, regardless of type, tend to be a necessary evil. Engineers are fully aware that a properly designed and manufactured one-piece structure—be it a forging, a casting, an extrusion or a composite layup—will be stronger, stiffer, lighter and more resistant to fatigue than an equivalent structure made up of several joined components. The reason for the relative weakness of joined structures is that stress cannot flow efficiently across joints. Unfortunately, at least from the viewpoint of the purist, unitary structures are often impractical from standpoints of cost, inspection, maintenance, repair and transport.

Most engineering structures and objects are therefore made up of several distinct elements that are joined by welding, by bonding or by some sort of mechanical fasteners. Because there are a number of excellent books dealing with the art of welding we are not going to worry about welding in this book. The relatively new and exciting science—or black art—of adhesive bonding I am going to leave for another book.

There are a great many subcategories of mechanical fasteners, but to my mind there are only two basic structural groups: threaded fasteners and rivets. The family of threaded fasteners includes bolts, studs, machine screws, metal screws and their associated nuts, inserts, washers, locking devices and thread locking chemicals. Rivets are divided into blind rivets or buck rivets and subdivided into solid and hollow.

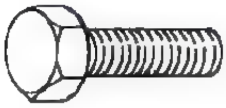
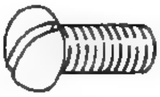
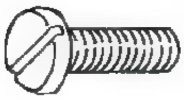
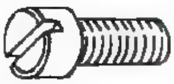
Each group has its rightful place in the technology of assembly. Threaded fasteners have several advantages over rivets: First, they are considerably stronger; second, they can be safely stressed in tension as well as in shear, whereas most rivets can only be safely stressed in shear; third, they can be used to fasten considerably thicker sections of material than rivets; and fourth, they allow easy



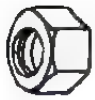
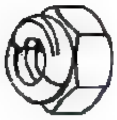
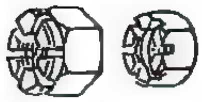
The helix.

The chambered nautilus shell, fabled inspiration for the helical screw thread.

Types of bolts and machine screws

	American usage	British usage
	Hex head bolt	Bolt
	Set screw Hex head cap screw	Setscrew
	Flat head screw	Countersunk head screw
	Oval head screw	Raised countersunk head screw
	Round head screw	Round head screw
	Pan head screw	Pan head screw
	Cheese head screw	Cheese head screw
	Slotted set screw Grub screw	Grub screw
	Allen bolt Socket head bolt Internal wrenching bolt	Cap head
	Button head screw	Button head screw
	Socket head shoulder screw	Shoulder screw
	Socket set screw	Socket set screw

Types of nuts and washers

	American usage	British usage
	Hexagon nut	Full nut
	Low hex nut Shear nut Jam nut	Lock nut
	Nylon ring elastic stop nut	Self-locking nut with Nylon insert
		Aerotight stiff nut
	Castellated nut Slotted hex nut	Castle nut Slotted nut
	Wing nut	Wing nut
	Acorn nut Dome nut	Dome nut
	Flat washer (plain and chamfered)	Stamped washer (plain and chamfered)
	Lock washer (split)	Spring washer (single coil)
	Lock washer (double split)	Spring washer (double coil)
	Star washer (external teeth)	Shakeproof washer (external teeth)
	Star washer (internal teeth)	Shakeproof washer (internal teeth)

disassembly or inspection by simple unbolting where rivets would have to be drilled out and replaced.

We will attack the threaded fastener first.

The helix

The familiar screw thread is based on a mathematical curve termed a helix. The helix is properly defined as the path described by a point that travels along the circumference of a circle at the same time that the circle advances along a line drawn through its center. Like most scientific definitions, this one is pretty confusing without a picture. My representation of a helix is shown here. The drawing should clarify the point somewhat. To clarify it completely, take a length of broomstick or other round rod and progressively wind a length of string around its outside diameter. The string forms a helix. If each turn of string just touches the turns ahead and behind, it is termed a tight helix. Greater spacing between the turns produces a loose helix. The string is called the thread of the helix and is normally cut into the cylinder rather than being added to it. In the case of the broomstick, you have created a male thread because, like male plumbing, it is on the outside. If the helix spirals clockwise (that is, turns to the right as it advances) it is called a right-hand thread. If the helix spirals counterclockwise it forms a left-hand thread. Like people, most (but not all) threads are right-handed.

In Greek mythology, the first engineer was Daedalus. In addition to other things, Daedalus is the hero of one of my all-time favorite books, *The Daedalus Testament* by a British sculptor/writer named Michael Ayrton. (If anyone knows how I can get my hands on a copy of this book, call me collect!) Anyway, we know about Daedalus because of the fabulous wings of wax and feathers with which he and his son, Icarus, escaped from Crete after he had constructed the Minoan labryinth and gotten himself into all sorts of trouble with a lady and a bull.

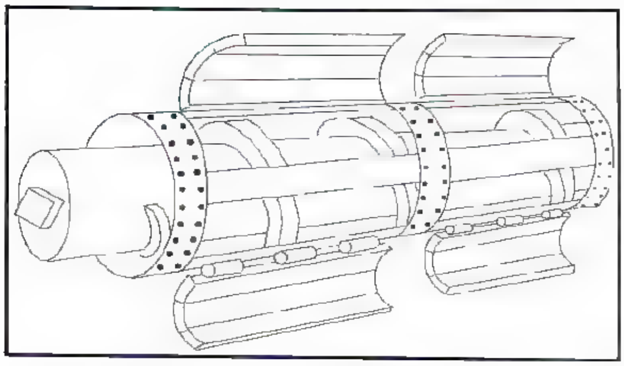
Icarus, being a son, ignored his father's flight plan and flew too close to the sun. When the wing feathers fell out of the melting wax, Icarus became the first recorded flight fatality—as well as the prototype son of an engineer.

What is not so well known is that before Daedalus left his native Greece, he was inspired by the shell of a chambered nautilus to develop the helix. As a point of interest he was also inspired by the skeleton of a fish to develop the saw blade. Clever fellow! To my mind this is indicative of the nature of the engineer: we are not innovators, we are developers.

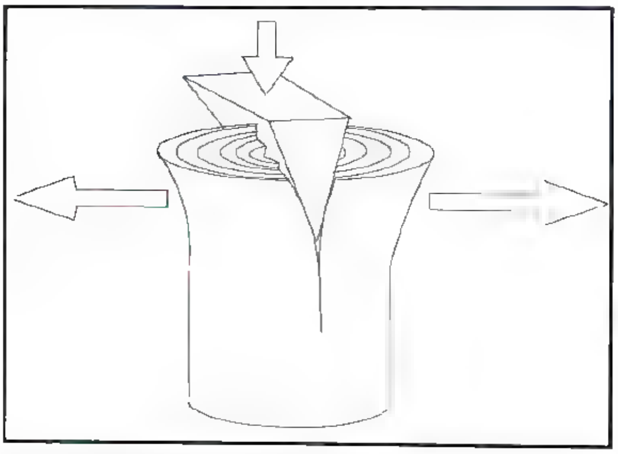
Be that as it may, history tells us that a later Greek, Archimedes, took Daedalus' helix and from it developed a water pump—the Archimedes screw. While we have developed more efficient pumps, we still use the Archimedes screw to dig post holes and to move grain and similar dry stores. The Archimedes screw and the Roman developments of it were handmade from wood. Lack of metallurgical and manufacturing technology prevented Daedalus and his contemporaries from actually *doing* much with the helix. In fact, the same lacks prevented their descendents from doing much more with it for a very long time.

We use screws for several purposes besides that of Archimedes. We use them to precisely control the motion of parts of various mechanisms, as in the lead/traverse screws of machine tools or the flaps/spoilers of large aircraft. We use them to vary the dimensions of and to locate parts of structural members, as in the rod end bearing and the turnbuckle. But mostly we use them to fasten parts together, as in the wood screw and the nut and bolt.

No matter what the application of a screw, there must be a matching and mating female threaded part to go with the male helix. In some cases—including those of the wood screw, the lag bolt and the family of self-tapping metal screws—the material to be fastened is either soft enough or thin enough for the male screw itself to form



The first known practical application of the helix—Archimedes screw, used here as a water pump.



The wood-splitting wedge at work.

matching threads in the material. When, however, we are talking about threaded fasteners used to hold metallic structures together, we are talking about a separate preformed female thread. With the bolt and the machine screw, the matching part is either a matching female thread cut into a hole drilled into one of the parts to be fastened or a removable nut. A nut is merely a piece of (usually) hexagonal stock that has been drilled to a predetermined inside diameter and has a matching female thread cut on its internal surface.

Thread Talk One: The wedge

When used on a fastener, the function of the helix that is of most interest to us is its wedging action. We have all used wedges, either to force two bodies apart (as in splitting wood) or as a lowly doorstop. The action of the wedge is simple and effective: when you exert a force on the short edge, the slope of the long side multiplies that force by the mechanical advantage of the slope and applies it in the direction normal (that is, perpendicular) to the direction of the applied force.

The screw thread is, in actuality, nothing more than a tapered wedge that has been wrapped around a cylinder. In effect, the force that we apply in turning the screw pushes against the short edge of the wedge. The upper surface of the male thread, as it engages the lower surface of the matching nut thread, acts like the angled long side of the wedge. Thus the force exerted in turning the screw is multiplied by a mechanical advantage proportional to the slope of the thread angle. This is how the scissors or bottle jack in the truck of your car allows you to lift your car off the ground with minimal effort. It is also how the threaded fastener develops large clamping forces.

When it comes to the development of clamping force, we must understand two distinctive characteristics of the helical thread: First, when we turn a male helix through a female mating part, the helix must advance as it turns; and second, when the threads advance to the point where the underside of the fastener head bottoms against a work surface, further turning will bring the wedging action into play. The head stops, the thread continues to advance and the body of the bolt must stretch (or strain) in reaction to the tensile stress developed in the bolt. When we stop turning the head of the bolt, the strain and the residual stress remain and generate a clamping force proportional to the residual stress.

That is the principle of the screw thread, and it is just that simple. The male part must advance as it turns through the female part and when the head bottoms, further tightening will produce a tensile stress and a clamping force in the bolt. The bolt is preloaded in tension. We use threaded fasteners to clamp parts together into structures. The principle may be simple, but the practice is not. Before I go any further, it is time for more undisguised definitions.

Thread Talk Two: Parts of the thread

Referring to the drawing, we have a few more terms to define:

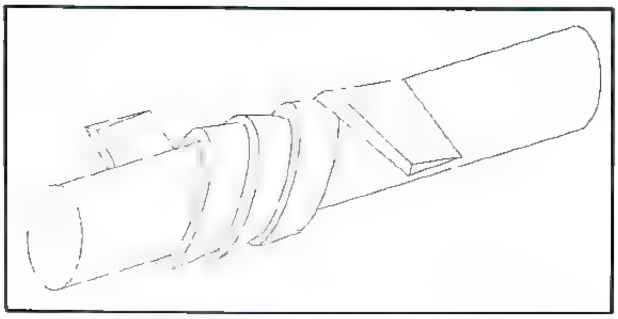
A. Form of thread—the profile of a thread, projected in the axial plane, for one full pitch.

B. Major diameter—the largest diameter of a screw thread.

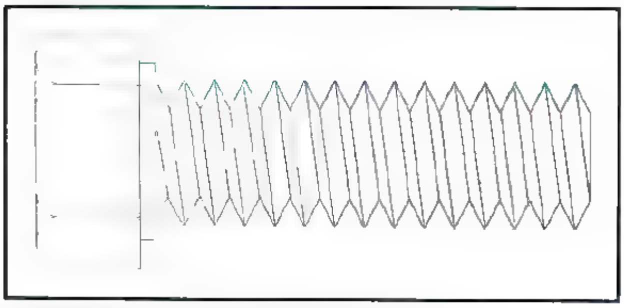
C. Minor diameter—the smallest diameter of a screw thread.

D. Pitch—the linear distance from a point on the thread to a corresponding point on the next thread—measured parallel to the axis of the thread.

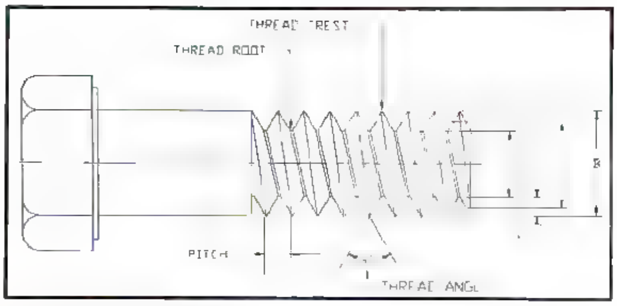
E. Pitch diameter—technically, the pitch diameter of a straight screw thread is the diameter of an imaginary cylinder that would pass through the



The helical thread as a wedge—theory.



The helical screw as a wedge—practice.



The parts of the thread.

profiles of threads at such points as to make the width of the thread grooves equal to one half of the pitch. For practical purposes, we can say that the pitch diameter is about halfway between the minor and major diameters of the thread.

F. Lead—the linear distance that a point on a screw thread will advance axially in one revolution. It is equal to the pitch of the screw.

G. Thread angle—the included angle between adjacent flanks of a screw thread measured in the axial plane.

H. Pressure flank—the load bearing flank of a thread. On a male thread the pressure flank is the flank toward the work face. On a female thread it is the flank away from the work face.

I. Thread height—the distance, measured perpendicular to the axis, between the minor diameter and the major diameter. Thread height is also called thread depth and depth of thread.

J. Thread root—the surface of the thread that joins the flanks of adjacent threads and is immediately adjacent to the cylinder from which the thread projects. In other words, the bottom or the valley of the thread.

K. Thread crest—the surface of the thread that joins the flanks of the thread and is farthest from the cylinder from which the thread projects. In other words, the top or the peak of the thread.

L. Stress area—the assumed diameter of an externally threaded part. This diameter is used in the computation of tensile strength, shear strength and fatigue life. The stress area is the cross-sectional area of the minor diameter of the thread—with allowance for manufacturing tolerances.

Parts of the bolt

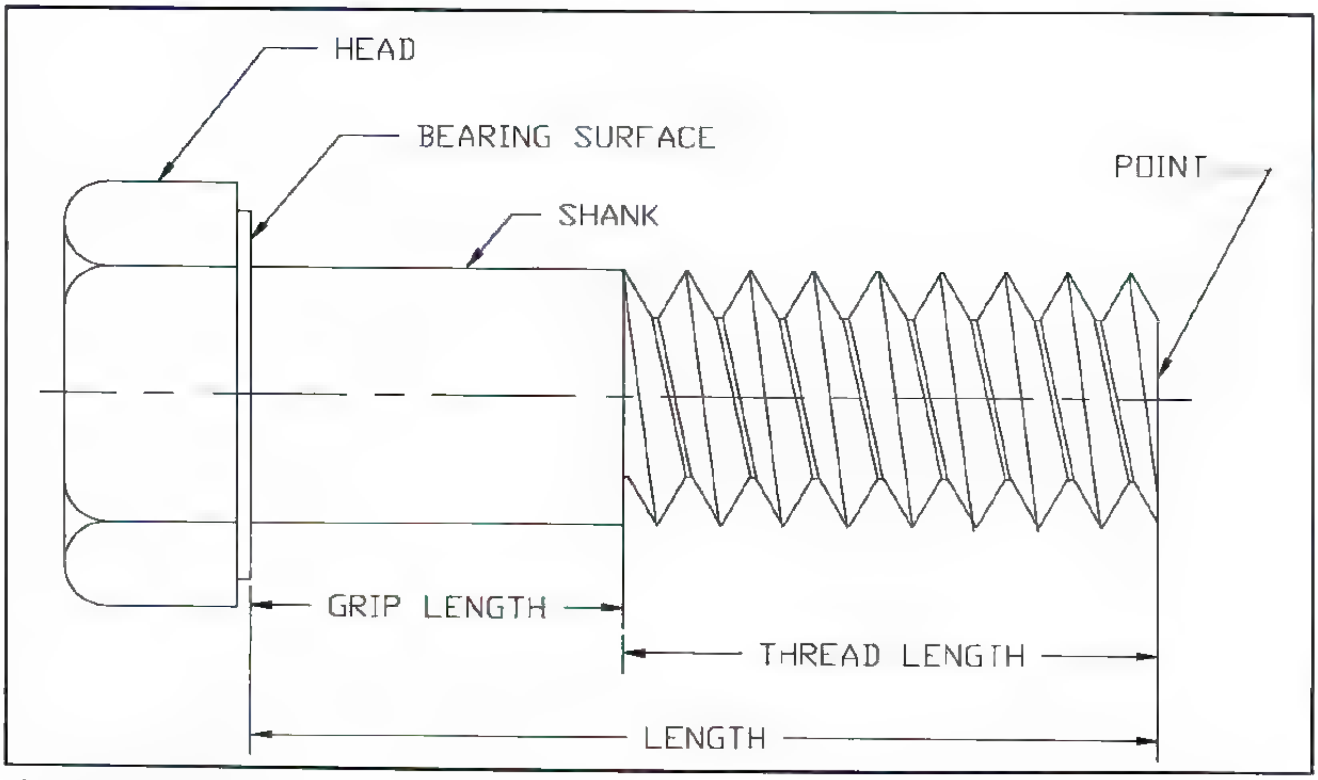
There are bolts and there are screws. Technically, a bolt is held in place by a nut while a screw is threaded into one part of the element to be clamped. In operation, a bolt is held stationary and tightened by turning the nut. A screw, machine screw or cap screw, on the other hand, is tightened by turning the head. So when it comes to distinguishing between bolts and machine screws, the application determines the terminology.

Referring to the diagram shown here, the major parts of a bolt (or machine screw) are:

A. Bolt head—the enlarged shape that is formed on one end of the bolt to provide a bearing surface and a method of turning (or holding) the bolt.

B. Bearing surface—the supporting or locating surface of a fastener with respect to the part that it fastens. In a bolt, the bearing surface is always the underside of the head and it should be circular in shape, as well as machined true and normal to the bolt axis. The bolt is loaded through its bearing surface.

C. Point—the extreme end of the threaded portion of a bolt.



The parts of the bolt.

D. Shank—the cylindrical part of a bolt that extends from the underside of the head to the point.

E. Length—the distance between the bearing surface of the head and the extreme point. It is measured parallel to the axis of the bolt.

F. Grip length—the length of the unthreaded portion of the bolt shank. It is measured from the underside of the bearing surface to the starting thread.

G. Thread length—the length of the threaded portion of the bolt. With all commercial and aerospace bolts, threaded length is a fixed function of bolt diameter.

Thread Talk Three: Types of threads

The helical screw thread is a versatile device that lends itself to many uses. There are several different specialized types of threads in normal use. For example, the Acme thread is used to produce traversing motions in machine tools and actuators. This is the square form thread that you see on the traversing rods of lathes and milling machines and on the flap actuators of commercial aircraft.

A second example is the buttress thread form which is designed to be capable of withstanding high unidirectional stresses parallel to the axis of the thread. You see the buttress thread on hose clamps.

Because they form their own matching female threads, wood screws and self-tapping metal screws have no need for standardization. They are highly specialized to suit different materials and assembly methods.

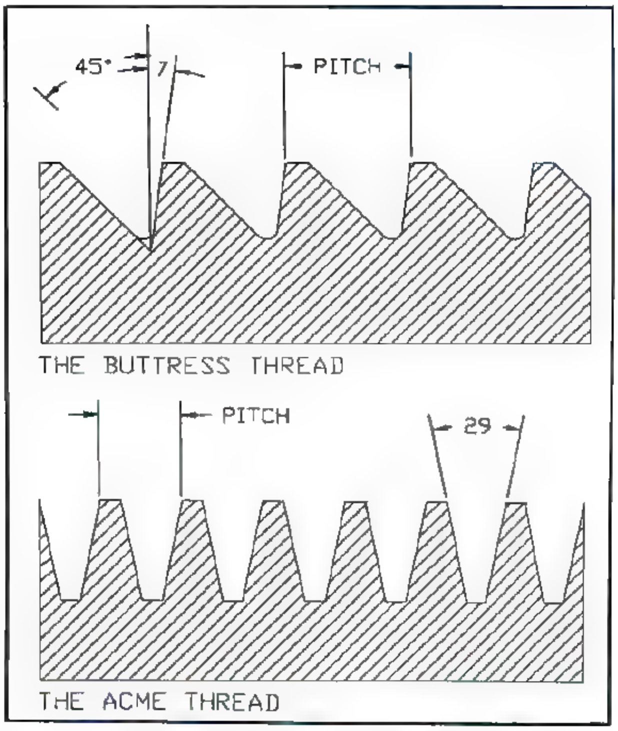
We are not going to concern ourselves with any of the highly specialized thread forms for soft materials, but will concentrate on the thread forms used in fastening the materials that we are interested in, such as structural metals and composites. For those of you who are interested in the fastening of high-tech wooden structures, I recommend Alex Strojnik's series of low-power laminar flow aircraft books, Tony Bingelis's *The Sport Plane Builder* and the Experimental Aircraft Association's *Aircraft Maintenance Manual.* If you are interested in high-speed assembly through sheet-metal screws and the like, I recommend the annual *Assembly Technology Buyer's Guide* from Hitchcock Publishing Company (see appendices for addresses), and *Machine Design*'s annual or biannual fastening reference issue. For the rest of us, it is time to learn more about the screw threads used in our world.

Fine versus coarse

The first thing you will notice about fastener threads is that, regardless of the country of origin, they come in two varieties: fine pitch and coarse pitch. Each has its place in the overall scheme of fastening, although there is no measurable difference in fatigue resistance between coarse threads and fine threads as such. The basic rule for selection is: When the female thread will be weaker than the male thread, use a coarse thread pitch. The reason is that the smaller the minor diameter of a female threaded part, the greater will be the thread area, the stronger will be the static strength and the higher will be the fatigue limit of the part. Conversely, the larger the minor diameter of a male threaded part, the greater will be the stress area and the stronger and more fatigue resistant the bolt or stud will be.

For this reason the steel bolts and studs that thread into castings such as engine blocks, cylinder heads, gearboxes, transaxles and the like are always coarse threaded on the end that goes into the casting. Also invariably, the end of the stud that receives the nut is provided with a fine thread. Of course, the length of engaged coarse thread is considerably longer than the engaged fine thread. In this way the designer ends up with the best of both worlds.

Coarse threads are also faster to assemble and take apart. This feature is of little interest in the world of high-performance vehicles—whether they perform on land, sea or air. In the world of industry, however, time is money and very often weight and bulk are not critical. Speed of assembly is the major reason that so many automotive and industrial fasteners feature coarse threads. Strength and weight are the reasons that almost all aerospace fasteners feature fine threads. Except when I am threading into a casting, I do not use coarse threaded fasteners.



The buttress thread and the Acme thread.

Thread Talk Four: Standardization

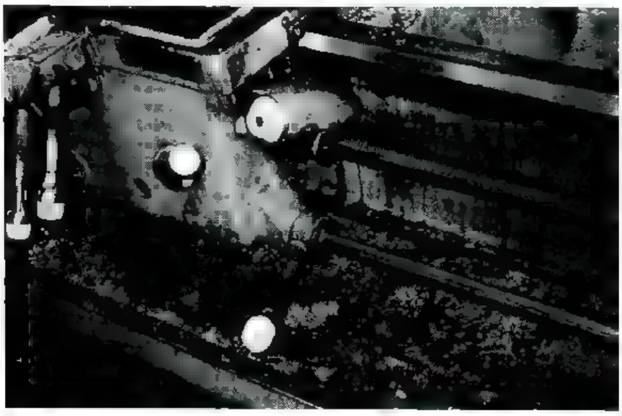
The screw thread has been around for a long time; threaded fasteners have been in common use for over a century. As you might expect, in the beginning every manufacturer made threads to suit his individual purposes and preferences. It didn't take long for the industrialists to realize that some sort of standardization was necessary. National standardization within specific industries was common by the late nineteenth century. In fact, Sir Joseph Whitworth introduced in 1841 the standardized 55 degree radiused root thread that bears his name and was the standard of industry for more than a century. As a point of interest, Whitworth's partner, Maudslay, developed a bench micrometer accurate to 0.0001 in. in 1835, and Whitworth had a comparator accurate to 0.000001 in. by 1835.

It took the international experience of World War II to make it obvious that, at least among military allies, some sort of international standards should be established for screw threads (as well as for weapon calibers). As a prime example, the most successful American fighter aircraft of World War II was the P-51 Mustang (this statement is guaranteed to start a good argument in any hangar, ready room or flight lounge in the world). As designed, with the US-manufactured Allison engine, it was an underpowered dog. The United States sold some to the British who instantly installed the Rolls-Royce Merlin engine. The engine transformed the Mustang into the dominant fighter aircraft of the European theatre of war. So we started manufacturing the engine under license in the United States. Naturally the engine was full of Whitworth and British Standard threads which were totally unknown to US industry or to the US Army Air Corps. Oops! At least the US Mustang's initial service was in England where the engine was born.

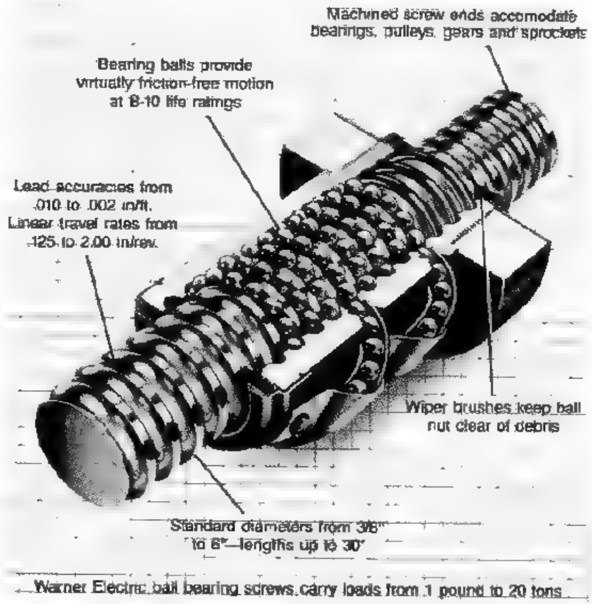
In 1948 the United States, Great Britain and Canada agreed to establish specifications for a standard series of unified screw threads featuring an included thread angle of 60 degrees. The series were designated as the unified fine (UNF) threaded series and the unified coarse (UNC) threaded series. These series replace and supercede the old American National Fine (NF) and American National Coarse (NC) series. It was agreed that thread fits would be called out in ascending order of tightness from loose to interference as classes one–five, respectively, with the subdesignation A identifying external threads and B denoting internal threads.

Specialized threads

There are some other thread forms in specialized use. While we are not liable to run into them, I will mention the least uncommon ones. Sometimes in the nineteenth century, the people who built bridges and the like determined that when threaded fasteners got big—as in over 1 inch in diameter—it was both convenient and structurally sound to call out a uniform pitch, regardless of fastener diameter. At one point we had the 8N or 8 pitch series for use on bolts and studs over 1 in. in diameter when coarse threads were desired, the 12N or 12 pitch series for bolts and studs over 1½ in. diameter where medium pitch threads were desired, and the 16N or 16 pitch series for fine threaded diameters over 2 in., as well as the NEF series of very fine pitch threads for use on thin-wall tubing, coupling nuts and so on. These have all been superceded by constant pitch additions to the unified thread series such as 4UN, 6UN, 8UN, 12UN, 16UN, 20UN, 28UN and 32UN for bolts, studs and nuts, and 27UN for thin-walled tubing and so forth.



The Acme thread as an actuator.



The Warner Electric ball bearing screw.

Thread Talk Five: Fit classifications

Class #1 is a loose fit designed to provide quick assembly and disassembly for lightly stressed fasteners used to join components that are taken apart frequently. So far as I know, I have never seen a Class 1 thread.

Class #2 is the general application fit. Most SAE-graded hardware is made to this specification. These fasteners are suitable for noncritical applications at medium stress levels.

Class #3 is a very close fit. It is, in fact, the closest fit available without interference. Good socket-head cap screws and most aircraft and aerospace bolts and nuts are manufactured to this specification.

Class #4 was a selective assembly fit that is no longer in use.

Class #5 is an interference fit designed for the permanent installation of studs in hard materials. Since I do not believe that a threaded fastener should ever be installed on a forever basis (no faith), I do not use Class 5 fits.

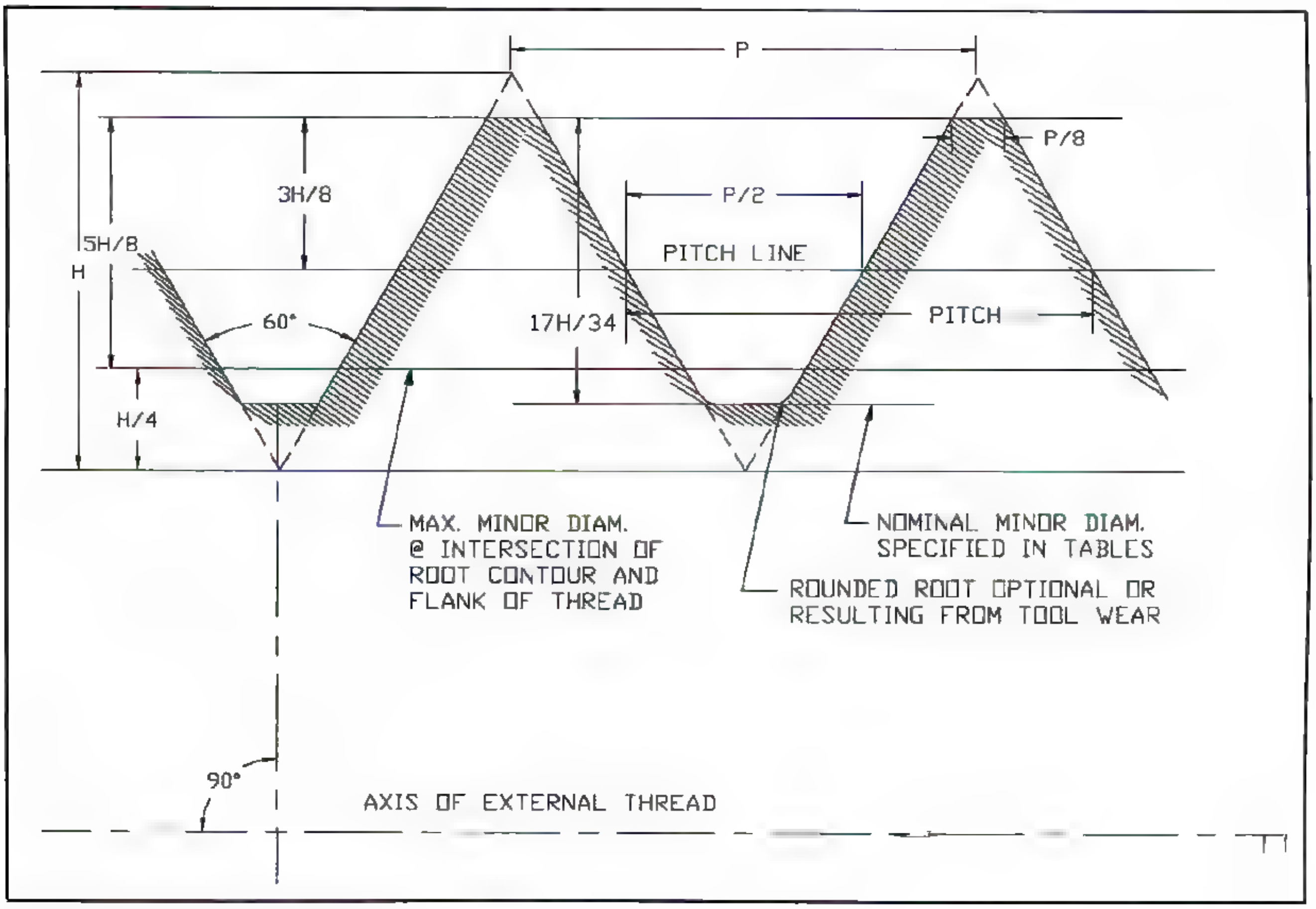
Tap fits

Internal threads are cut by taps. Good taps are ground to specific size limits in accordance with standards set up by the American National Standards Institute (ANSI). The part number etched on the tap is coded to indicate the limits. As an example, a tap marked ⅜x24 UNF GH3 LH designates the following:

- ⅜x24 UNF: This tap will produce a thread ⅜ in. in diameter with twenty-four threads per inch. The thread form will conform to the standards set forth in the Unified Fine Thread specifications published by the American National Standards Institute.
- G: The cutting surfaces were finished by grinding.
- H: Tap diameter is larger than the called-out pitch diameter. If this call out were L instead of H, the tap diameter would be lower than the called-out pitch diameter.
- 3: This call out specifies the difference between tap diameter and specified pitch diameter in increments of 0.0005 in. In this case the difference is 3x0.0005 in. or 0.0015 in. larger than specification.
- LH: The final suffix LH indicates that the tap will cut a left-handed thread.

Thread Talk Six: Rolled versus cut threads

Male threads can be manufactured either by cutting (on a lathe or with a thread die) or by roll-



The UNF thread form (external thread).

ing. In the thread rolling process, the cylindrical bolt (or stud) stock is passed between rotating grooved rollers under extreme pressure. The thread is cold formed by mechanical displacement of metal into the grooves in the rollers. The process is not dissimilar to forging. The grain structure of the metal actually follows the contour of the threads. The finished thread surface is not only smooth, it is actually burnished by the rollers. As you would expect, the resulting threads are both strong and resistant to fatigue. The only practical way to form female threads is to cut them, so female threads are nowhere near as strong or fatigue resistant as male threads.

One of the most unpopular proclamations that I made in *Prepare to Win* was that either die cutting or lathe cutting threads onto a bolt is a crime against nature. I have been attacked by just about every machinist that I know. They state that any competent machinist can turn good threads onto any piece of bolt stock that I can come up with. No argument! I never stated that geometrically good threads could not be turned on a lathe. They certainly can be—assuming that the operator knows what he is doing, uses a properly shaped and sharpened tool and the right speeds for the thread and the metal. Trouble is, no matter how competent and careful the operator may be, cutting threads across the grain of the metal interrupts the flow of stress through the bolt. This means that not even the best lathe-turned threads are going to be first, as strong as rolled threads, and second, as resistant to fatigue as rolled threads. Turning involves tearing metal and leaves rough spots which form perfect stress raisers.

To the best of my knowledge, all commercial bolts and studs that we are liable to come across are manufactured with rolled threads. Only very large threads are commercially cut anymore. As a point of interest, this fact has at least as much to do with economics as with a desire to produce strong threads. The first thread-rolling machines were developed in the 1870s because, even then, it cost too much to cut threads in mass production.

As you might expect, however, there are various ways of rolling threads. The results tend to vary directly with the cost of the equipment and the process. The right way is to cold roll the threads in a single pass through polished rollers that have radiused thread roots. Threads should be rolled after the bolt or stud blank has been heat treated. Threads rolled before heat treatment are not much better than cut threads. There are several reasons for this.

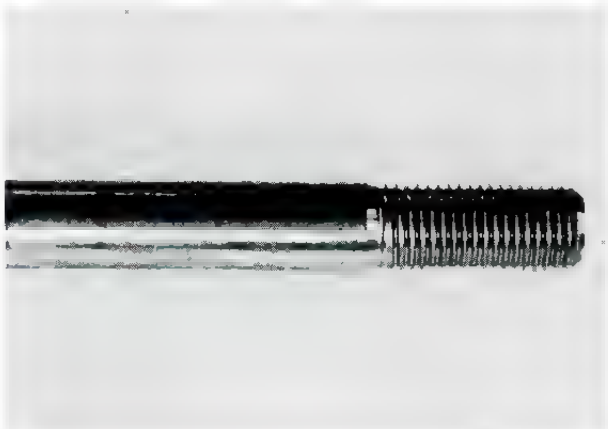
First, the process of rolling threads creates high compressive stresses in the root of the thread. The effect is similar to that produced by shot peening but many times greater. This compressive layer increases the fatigue life of the fastener by an order of magnitude. If the fastener is heat treated after thread rolling, the normalizing process removes the compressive layer.

Second, heat treatment inevitably results in some physical distortion of the fastener blank, which must be machined and ground true before it can be threaded. Rolling the thread onto the trued blank after heat treatment ensures that the thread will be perfectly coaxial with the bolt and normal to the bearing surface of the bolt head.

Third, rolling the thread on an already heat treated blank will remove by cold working any imperfections resulting from the heat treat process.

Real high-strength bolts have the threads rolled after the heat treat, and to hell with the cost. I don't know exactly where the dividing line in the industry lies, but, for example, all of the NAS bolts that I have checked specs on specify UNJF threads

Comparison of different price and quality levels of rolled threads. From right, A Cosworth rod bolt, NDS bolt, SAE Grade 8 bolt, SAE grade 5 bolt, and a junk bolt. Roy Kiesling

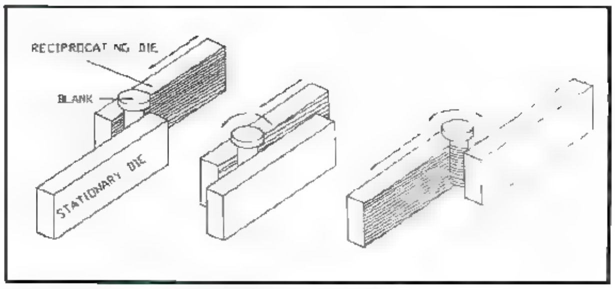


A quality rolled thread. Roy Kiesling

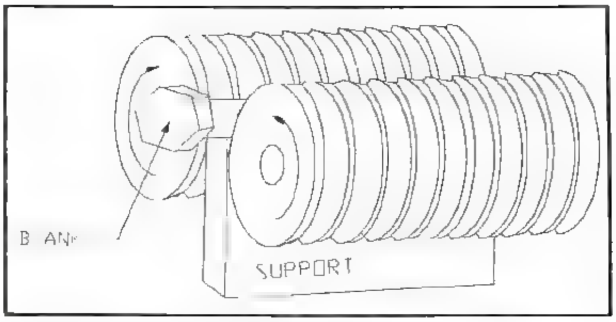
to be rolled after heat treat—as do all of the socket-headed cap screws that feature UNR threads. A quick comparison by naked eye between the threads on a super-whatever bolt and an AN/MS/NAS bolt will illustrate my point, as will a superficial glance at the Taiwanese and Korean counterfeits.

Microphoto of cut thread. Note rough radius between shank and head. Roy Kiesling

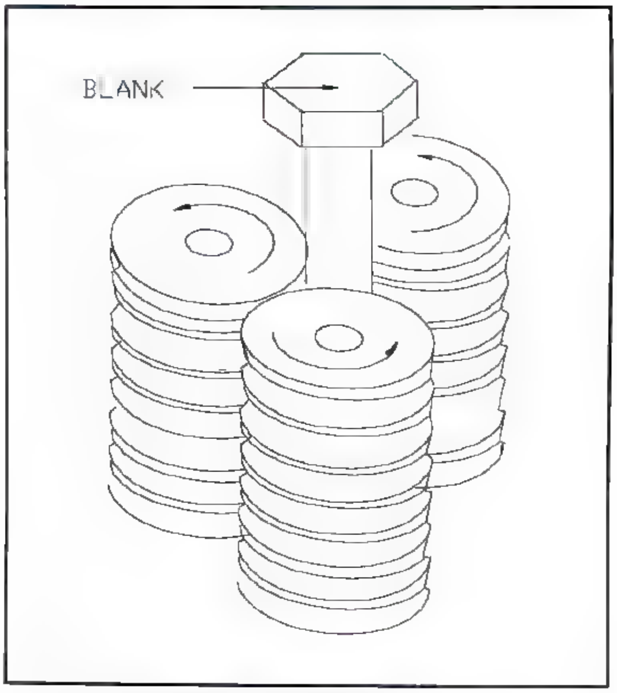
So we not only have to guard against the well meaning but uninformed person who decides to modify an existing bolt—or to make one—but also against the cost-cutting bean counters who run the industrial fastener corporations. One problem here is that the SAE has no practical way to enforce their



Thread rolling with reciprocating dies.



Thread rolling with two cylindrical dies.



Thread rolling with three cylindrical dies.

standards. A second is that purchasing agents are not engineers and are rarely knowledgeable about engineering. There is no way on God's green earth that anyone is going to cut even a decent thread with a thread die—and this includes just "cutting a couple of more threads onto this here bolt."

Thread Talk Seven: Radiused thread forms

The Unified Thread Standards stress the desirability of a rounded or radiused root shape for the external (bolt) thread. Unfortunately, in what seems to me to be remarkable compromise, the standards permit flat thread roots when "new tools" are used to produce the threads. God (and decades of experience) tells us that, for maximum strength and especially for maximum resistance to fatigue, the root of the thread should be radiused. There are a few alternative radiused thread forms currently used in both conventional industry and in aerospace.

The UNJ thread was developed in aerospace when it became obvious that a thread form offering more resistance to fatigue was required. The UNJ specifications call out a thread root radius of 0.150 to 0.180 times the pitch length. The UNJ thread is used on most aerospace bolts with an ultimate tensile strength of 160,000 psi and above. It first became known outside the aerospace industry as the J thread on the bolts that Standard Pressed Steel supplied to solve the connecting rod breakage problem on Ford Motor Company's four-cam Indianapolis engine in the mid 1960s. Where the regulations permit them, every racing engine builder in their right mind still uses SPS bolts with the UNJ thread for connecting rod bolts. Standard nuts and taps suit both UNR and UNJ threads which, except for the included angle of the thread, are remarkably similar in form to the British Whitworth thread, which was abandoned in favor of UNF and UNJ back in 1948.

The UNR thread is used in general industry when increased fatigue life is required. Virtually all good-quality socket-headed cap screws now utilize this superior thread. A thread root radius of 0.108 to 0.144 times the pitch length is called out and the bolts are available in both fine (UNRF) and coarse (UNRC) threads.

Thread Talk Eight: Metric threads

Those of you who have read my previous books will be aware that I am a fan of the metric system and favor its adoption in the United States. It is not that I don't *like* our inch system, but having lived and worked in a number of other countries, I am aware that the metric system is easier to use (any damned fool can multiply or divide by ten). It is not difficult to learn or adapt to. I managed, and I am not noted for patience, tolerance or willingness to change something that works.

More important, however, is the fact that metrication is necessary for our survival as an industrial nation. Everyone agrees that our nation must export both industrial products and technology to survive. I maintain that the only way we can hope to export our products is to make them better than our competitors do, not cheaper.

I further maintain that we must export products tailored to the measuring systems of the intended users—and the rest of the world is not about to adopt *our* system. Since we export the same goods that we use, it makes no sense to manufacture two parallel product lines—one dimensioned in inches for the home market and the other in metric units for export. The cost in redundant tooling and labor would be ridiculous. In our refusal to metrify, we stand almost alone. The last time I bothered to check, the other countries not using the metric system were Burma and Papua New Guinea. At any rate, I will now stop preaching and get on with the book.

Least we think that the metric system is perfect, allow me to point out that currently there are at least four different metric thread systems in common automotive use. They are: JIS, DIN Standard, DIN Fine and ISO.

Fortunately, within most industries (including the automotive industry), metric threaded fasteners are now manufactured to conform to International Standards Organization (ISO) specifications which correspond to our American National Standards Institute, American Society for Testing and Materials (ASTM) and Society of Automotive Engineers (SAE) standards. The metric thread form is similar to that of UNF/UNC, although the depth of metric thread is slightly greater. The included thread angle is 60 degrees. Bolt heads are finished with washer bearing surfaces. Partially threaded bolts have thread lengths corresponding to the length of the bolt.

Up to 125 mm = 2 diameters + 6 mm
125 mm to 200 mm = 2 diameters + 12 mm
Over 200 mm = 2 diameters + 25 mm

Tensile strength is designated by a two- or three-digit marking on the bolt head. Common designations are 8.8, 9.8, 10.9 and 12.9. The digits to the left of the decimal point indicate 1/100 of the ultimate tensile of the bolt material in Newtons per square millimeter (N/mm). The second figure indicates the ratio of yield strength to ultimate tensile strength.

A unit of stress 1 N/mm is equal to 145.037 psi, so the conversion goes as follows.

ISO marking	Minimum ultimate tensile strength	Minimum yield tensile strength
8.8	116,000 psi	92,800 psi
9.8	130,000 psi	104,000 psi

THREAD PITCH					ACROSS HEX FLATS		
DIAM	I.S.O.	J.I.S.	DIN STD.	DIN. FINE	I.S.O.	DIN	J.I.S.
3mm	0.5	0.6	0.5	0.35	5.5mm	5.5mm	6mm
4mm	0.7	0.75	0.7	0.5	7	7	8
5mm	0.8	0.9	0.8	0.5	8	8	9
6mm	1.0	1.0	1.0	0.75	10	10	10
7mm	1.0	1.0	1.0	1.0	11	11	11
8mm	1.25	1.25	1.25	1.0	12	13	14
10mm	1.25	1.25	1.5	1.0	14	17	17
12mm	1.25	1.5	1.75	1.5	17	19	19
14mm	1.5	1.5	2.0	1.5	19	22	21
16mm	1.5	1.5	2.0	1.5	22	24	23

Metric thread forms in common use.

ISO marking	Minimum ultimate tensile strength	Minimum yield tensile strength
10.9	145,000 psi	130,000 psi
12.9	174,000 psi	157,000 psi

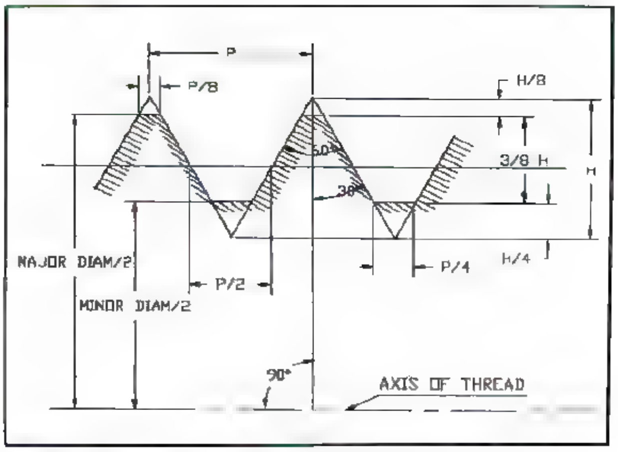
Metric bolts are available in both fine and coarse threads. In fact, there are two fine threaded metric series in common use, the 1 mm pitch series and the 1.25 mm pitch series and, yes, this does cause a lot of confusion. They are available as either hexagon-headed bolts or as socket-headed cap screws. Material specifications and markings are the same for either head style. A complete line of industrial-quality metric threaded fasteners is available, including button-head cap screws, hex- or socket-headed shoulder bolts, machine screws in all configurations, sheet metal screws, set screws, retaining rings and pins—along with matching nuts and washers of all descriptions.

It used to be that if we needed aerospace quality in a metric fastener we were in deep trouble, unless you knew someone at a European aerospace company like Dassault, Matra or Aerospaziale. Today we are still in trouble, but not as deeply. The need for metric-sized fasteners has extended to the aerospace industry. The National Aerospace Standards Committee of the Aerospace Industries Association of America has adopted a set of standards for metric fasteners designated NA Metric. SPS manufactures fasteners to this spec, and I suppose that most of the other biggies do as well. Any good aerospace fastener house should be able to obtain them, but on special order and not cheaply.

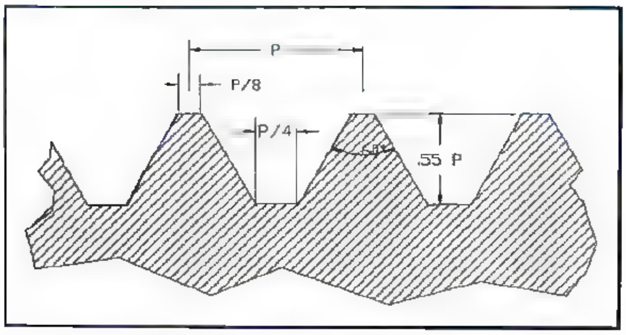
The NA metric thread form is based on the standard ISO metric standards and has a radiused root similar to the assassinated Whitworth and the current UNR and UNJ threads—leaving our flat-bottomed Unified Thread form standing alone in a world that apparently thinks that fatigue in fasteners is more important than we in the United States do.

Finding the metrics

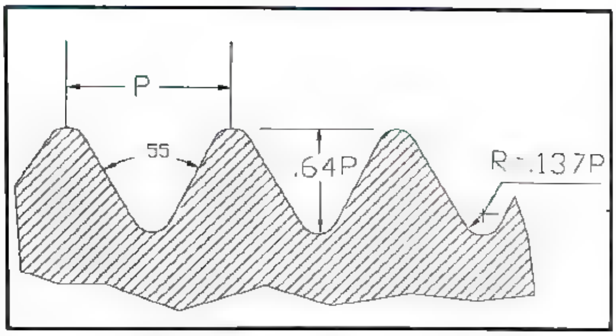
Even though our domestic auto industry now uses (very quietly) quite a few metric fasteners, and



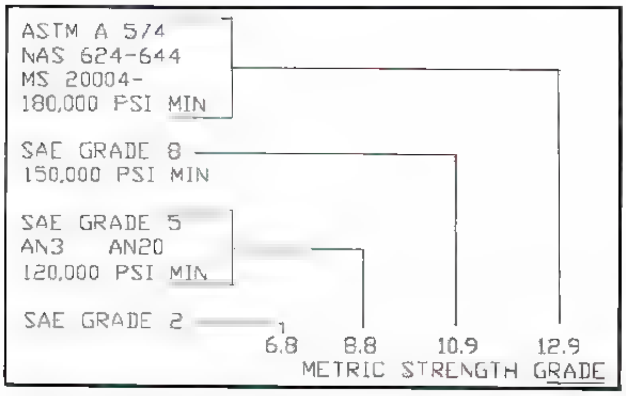
The ISO metric thread form (external thread).



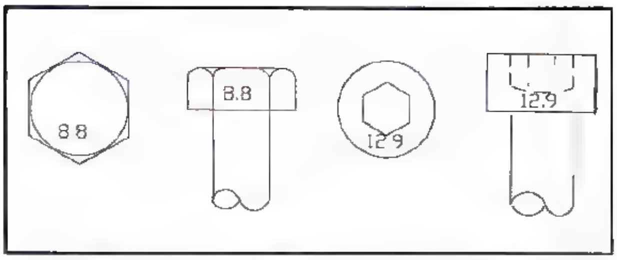
Comparison of thread forms: ISO metric thread form.



Comparison of thread forms: Whitworth thread form.



Standard United States bolt strength grades versus ISO metric bolt strength grades.



Metric bolt head markings. Markings are obligatory for property classes 4.6, 5.6 and all classes equal to or higher than 8.8. If low-carbon martensitic steels are used for property classes 8.8, 9.8 and 10.9, the symbol should be underlined as 8.8, 9.8 and 10.9

aerospace has written metric standards for threaded fasteners, it isn't easy to find metric hardware, let alone *good* metric hardware. The best sources are Global Metrics, Metric and Multistandard Components Corp. and Metrics Specialties Inc. (see appendices for addresses).

All of these companies stock industrial-quality hex-headed and internal wrenching bolts in all common metric diameters, thread pitches and strength levels as well as plain nuts, elastic stop nuts, all-metal lock nuts, and flat and lock washers. They also stock the copper-coated all-metal lock nuts that work magic for exhaust systems. Metric and Multistandard Components publishes an excellent catalog (#4011) complete with technical specifications and also stocks all of the British threaded fasteners. Global Metrics has a better catalog but without the technical information. None of these companies stocks aerospace-quality bolts.

In addition, any reasonably competent fastener house should be able to obtain any industrial metric fastener that you can name—in box lots (usually 100 pieces)—from relatively local distributors. You will have to know exactly what you want because the fastener house will not, especially when it comes to thread pitches and strength grades. We will get into the problem of counterfeit bolts later, but this is a good place to state that you should not touch bolts made in either Taiwan or Korea with a ten-foot pole. It is your right to know the origin of manufacture of anything that you purchase.

This is good information to have when you need a season's supply of something, but is not a lot of help when you need a couple of items in a hurry. You would do well to make the acquaintance of a good parts man in a Fiat, VW, Honda, Nissan or Toyota dealership. They have good hardware of all descriptions and some of it is stamped on the head with the standard ISO markings. A lot of it is not marked, though, and I would not use the unmarked hardware for anything critical. Japanese quality is every bit as good as the German and Italian and a damn sight less expensive. (In my opinion this applies to the cars as well as to the hardware.)

Function of the bolt

Now that we know something about threads, let's examine the function of the bolt—what the bolt actually does. Or, more accurately, what the bolt is designed to do. We sometimes stray a long way from the design function, but that will come later. No matter how they are loaded, bolts (or studs) are meant to clamp parts together—and that is *all* they are meant to do. They are most emphati-

Mechanical properties of metric bolts

Designation: Metric threads are identified by the letter M for the thread profile form. The letter M is followed by the nominal diameter in millimeters and the thread pitch separated by the sign X. (Example: M 10 X 1.25)

Property class: Symbols consist of two figures. The first indicates 1/100 of the nominal tensile strength in newtons/mm2 (1 N/mm2 = 145.037 psi). The second figure indicates ten times the ratio between nominal yield stress and nominal tensile strength. Multiplying these two figures together will give 1/10 of the nominal yield strength in N/mm2.

Property class	Material and treatment	Chemical composition limits %			
		C		P	S
		Min	Max	Max	Max
4.6, 4.8, 5.6, 5.8, 6.8	Low or medium carbon steel	—	.25	.05	.06
8.8	Medium carbon steel, quenched & tempered	.25	.55	.04	.05
9.8	Medium carbon steel, quenched & tempered	.25	.55	.04	.05
10.9	Medium carbon steel with additives such as Boron, Mn or Cr, quenched & tempered. Or alloy steel quenched & tempered	.20	.55	.04	.05
12.9	Alloy steel, quenched and tempered	.20	.50	.035	.05

The mechanical properties of metric bolts.

cally not meant to act as pivots, axles or fulcrums, nor are they meant to locate members in relation to each other.

In the context of fastening, location means to prevent parts from moving in relation to each other in the direction perpendicular to the axis of the threaded fasteners. In other words, threaded fasteners should not be used to prevent the clamped parts from sliding on each other. Location is the proper function of dowels, piloting diameters and the like. For example, bellhousings are typically located on engine blocks by dowels and then clamped in position by bolts. The same is true of flywheels—at least of well-designed flywheels. In the world of motor racing, transaxle gearcases are doweled onto the differential case and then clamped by studs, while differential side covers are located by concentric piloting diameters and clamped by studs as are the transaxles to the bellhousings. Bolts are meant to serve as clamps,

Mechancial property	Property class						
	4.8	5.6	6.8	8.8	9.8	10.9	12.9
Nominal tensile strength in psi X 1,000	58	77.5	87	116	130	145	174
Nominal yield stress in psi X 1,000	43.4	49	61	69	104	130	156
Rockwell Hardness B Scale	71	79	89				
Min C Scale				20	27	31	38
B Scale	95	95	95				
Max C Scale				30	36	39	44

The mechanical properties of metric bolts.

period. As clamps, they must remain tight under all kinds of loads and vibrations. A loose bolt not only makes a poor clamp, but as we will discover, it will fail quickly from fatigue.

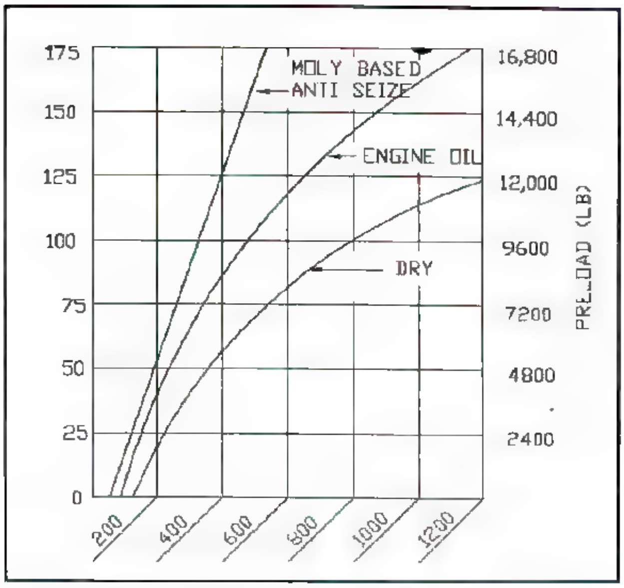
OK, so what actually keeps a properly designed and put-together bolted assembly tight? Let's first discuss some of the things that do *not* keep it tight. Safety wire does not keep it tight. Loctite does not keep it tight. Lock washers do not keep it tight. Even elastic stop nuts will not keep a bolted joint tight. Neither will castellated nuts and cotter pins.

In the final analysis, what keeps the bolted joint tight is the same force that clamps the parts together—the residual tensile stress set up in the bolt during the tightening process. All the rest is window dressing, largely employed to make up for the inescapable fact that, in the practical world, most bolts are improperly tightened. This last sentence is not a condemnation of standard engineering practice. Nor do I mean to say that all of our high-tech self-locking devices and chemicals exist mainly to make up for our inability to do something as simple as tightening a bolt. I am not saying that we are unable to or are too lazy to properly tighten the thing. I am saying that in the real world, it is not practical to properly clean and torque every bolt, every time—not in the world of racing cars or even aircraft. Further, since most of our applications involve only shear loads, in a great many cases it isn't even necessary.

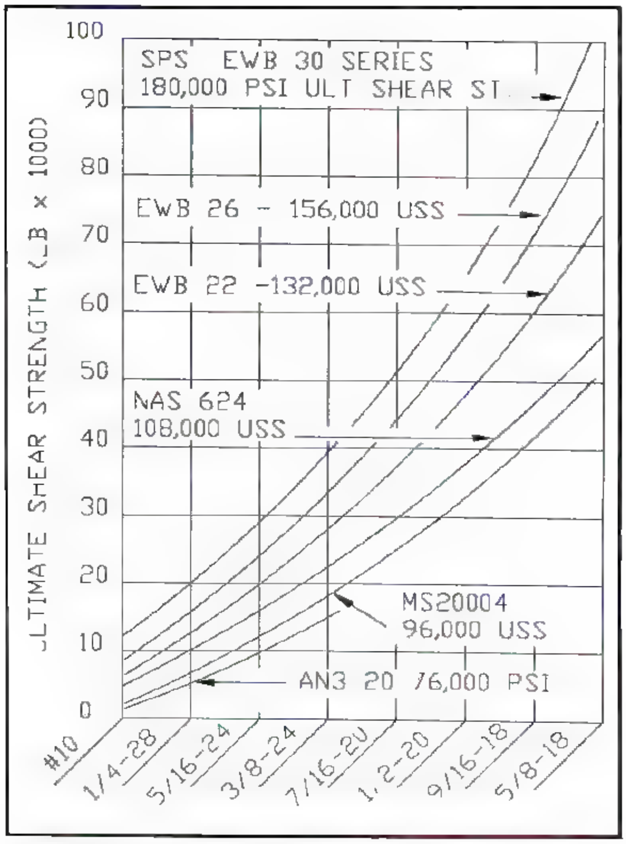
If the bolt is going to clamp parts together, sometimes against the opposition of considerable and cyclic force, then, when installed, the bolt must be internally stressed in tension—otherwise the clamping force will not exist. This is why we take considerable care in the tightening of our bolts. We may not realize it, but in tightening a bolt to a specified torque value we are actually stretching the bolt and loading it in tension to a predetermined level of stress. The English and the Australians correctly term our torque wrench a tension wrench. In fact, the level of installed tension (often termed preload) is more important to the strength and the fatigue resistance of the bolted assembly than is the ultimate strength of the bolt. A bolt that is preloaded or stretched to its designed level of residual stress will resist a given cyclic load for the maximum number of fatigue cycles. It will also provide maximum resistance to loosening from vibration. A bolt that is installed in an understressed condition will loosen under load and will then fail—either by rupture or by loss of clamping force. A bolt that is overtightened, on the other hand, will fail either during installation or prematurely from fatigue under cyclic stress.

Mechanics of tightening threaded fasteners—Or how tight is right

During tightening, any threaded fastener is subjected to two quite different stresses. First, the tension stress set up by the actual stretching (or strain) of the bolt as it is tightened. This stress does not begin to develop until after the bearing face of the bolt or nut has contacted the work face. Second, the shear stress (often mistakenly called torsional stress) that is caused by friction between the male and female threads and between the undersurface of the bolt head and the work face. This stress varies from virtually zero at the time that thread engagement begins to an extremely high value indeed at the end of the tightening operation.



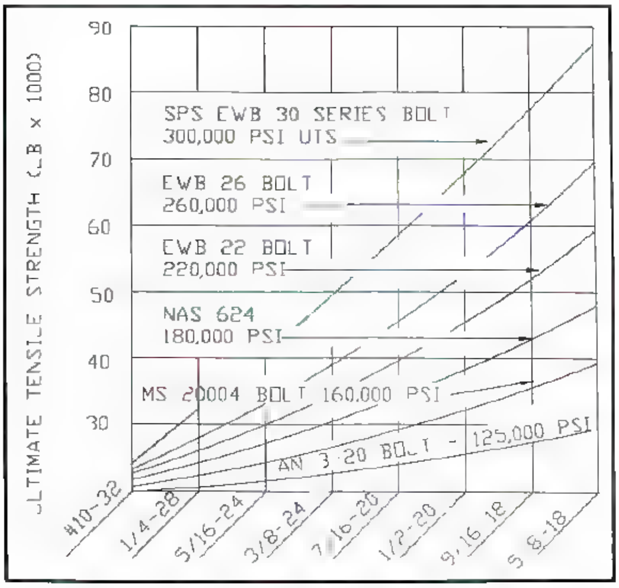
The effect of different lubricants on the torque required to achieve a given level of stress. (⅜ in., 160,000 psi UTS bolt.)



Ultimate shear strength comparison of various series of aerospace bolts by diameter.

The tension stress, produced by the strain or stretching of the bolt as it is tightened, is what we are looking for. This stress will remain in the bolt after the wrenches are removed and, as long as the total level of stress has remained within the elastic limits of the bolt, this residual stress will exert a strong clamping force on the assembly forever—assuming that the assembly is rigid.

The shear stress is undesirable but unavoidable. Fortunately (in most respects) it goes away almost as soon as we take the wrench off. It is undesirable because it is unpredictable. The amount of friction involved, and therefore the torque required to produce a given amount of tension, varies with the type of plating (if any) used on the threads, the cleanliness of the threads, the fit of



Ultimate tensile strengths of various series of aerospace bolts by diameter.

BOLT DIAMETER	TORQUE IN N*M	TORQUE IN LB/FT
4MM	3.7 N*m	2.5 LB/FT
6MM	13 N*m	10 LB/FT
8MM	31 N*m	23 LB/FT
10MM	62 N*m	46 LB/FT
12MM	108 N*m	79 LB/FT
16MM	268 N*m	198 LB/FT
20MM	524 N*m	390 LB/FT

Recommended tightening torque values for metric socket-head cap screws at 12.9 strength level using engine oil lube.

the threads and the type and amount (if any) of lubricant used. This makes it difficult to determine the amount of torque required to produce a given amount of stress and resulting strain within the fastener.

Just to give an indication of the percentages we are talking about, in the worst case, with dry and unplated threads, about fifty percent of the torque applied in tightening a common nut and bolt can be used up in overcoming the friction between the bearing surfaces and the work faces, while about forty percent can be used up in overcoming thread friction and ten percent actually contributes to preloading the bolt! This is why the multitude of torque tables are only a guide to proper tightening, not a bible. This is also the reason that, in critical applications (like connecting rod bolts in racing engines), it is necessary to physically measure the amount of strain in the bolt to determine when it has been tightened to the extent that will produce the desired residual stress. As we will see later, there are also strain-indicating washers and similar devices available to those of us who are both conscientious and wealthy.

The only good thing about the shear stress developed in the bolt by the friction of tightening is that it goes away almost as soon as the act of tightening is completed. When you stop turning the nut or the bolt, relative movement ceases. Without relative movement there is no friction—everything relaxes just a little bit and the shear stress evaporates. The tension stress remains. This is the reason that we always torque our tension bolts in steps—usually three steps—rather than just tightening the bolt to its final torque all at once. By going a little at a time we give the shear stress a chance to relax and we get a more accurate reading. It is also the reason why most mechanics click the torque wrench twice on the final tightening operation.

In the practical sense this means that, assuming that the bolt is strong enough for the application and that the joint is rigid, the highest level of stress that the bolt will ever experience is that reached during tightening. If the bolt doesn't fail during tightening, then it will not fail in service. This sounds good, but we all know from experience that it is simply not true. Either this is another case of theory being overcome by practicality, or are we overlooking something here. As usual, it is the latter and, in investigating the picture further we are going to start learning about the practical aspects of the bolted joint.

The residual tensile stress in the bolt is the force that will clamp the parts and lock the male and female threads together so that the assembly will not loosen in service. What actually happens is that, as the bolt stretches, the male threads elongate and the female threads compress. The result is an increasing interference condition that resists loosening due to vibration and so on. Unfortunately, as we have seen, a large percentage of the actual torque required to tighten any threaded fastener is used up in applying the shear stress necessary to overcome the friction generated by relative motion between the male and female threads as the fastener is tightened. This means that, while the calculation of the proper amount of preload is not all that difficult, its accurate measurement *is.*

Remember that the torque required to produce a given tensile stress varies with plating, lubrication (or lack of it), length of engaged thread and class of thread fit. As an example of the magnitude of what we are talking about, the chart shows the effect of different levels of lubrication on the tightening torque required to achieve various levels of installed stress in the same fastener.

What we are really looking for is a level of installed tensile stress that is just below the yield strength of the bolt material. Standard AN torque tables are usually compiled for plated fasteners, without lubrication. The standard torque table is for cadmium-plated AN-3 through AN-20 airframe bolts used with full-height AN-365 elastic stop nuts. Tension assemblies are different.

Critical tension assemblies subjected to high levels of cyclic stress such as cylinder heads, connecting rods, flywheels and the like require specialized fasteners and each such fastener usually has its own recommended torque value (and lubricant) to arrive at the correct level of installed residual stress.

These values are first calculated and then verified experimentally. The optimum level of installed stress is normally just below the yield stress of the bolt—between sixty and eighty percent of its ultimate tensile strength. Of course, as the ultimate strength of the bolt material is increased, the material becomes less ductile. The result is that the ultimate tensile strength and the yield strength grow closer together so that while the ideal preload spec for an AN-6 airframe bolt with a UTS of 125,000 psi will be at about seventy percent of the UTS or 87,500 psi, the call out for a 220,000 psi connecting rod bolt will be a considerably higher percentage—closer to eighty percent or 175,000 psi. This is not something that we have to be able to calculate. The specialized bolt manufacturers provide us with the information that we need. We must, however, understand why it is so; simply upgrading the strength of a bolt that has failed in service will not solve the problem unless we also increase the installed residual stress or preload.

Most clued-in engine builders measure the actual stretch of the individual connecting rod bolts rather than relying on an indicated torque value. In fact, we would install every tension bolt in this manner if we could. Since stretch measurement requires access to both ends of the bolt and most tension installations are blind, we cannot. In critical aerospace applications, stress-sensitive

washers and various types of stress-indicating bolts are used to ensure proper bolt preload. Only in these ways is it possible to take full advantage of the operating stress levels available in the current generation of high-strength bolts.

As a point of interest, we can now readily obtain bolts with ultimate tensile strengths of 220,000 psi, and yield strengths of 185,000 psi. Bolts are available (and nuts to match) up to 300,000 psi, but they are difficult for us mortals to obtain—and they have very little ductility. These charts show the ultimate tensile strengths, shear strengths and fatigue limits of various aerospace quality bolts. There *is* a difference.

Overtightening versus undertightening

Strange as it may seem at first, it is actually better to overtighten a bolt than to undertighten it! As an example, in a laboratory test, a 180,000 psi tension bolt with a shank diameter of ⅜ in. was prestressed to forty percent of UTS and subjected to a cyclic tension load of 12,000 lb. failed after 4,900 cycles. An identical bolt, prestressed to a level of 108,000 psi and subjected to the same cyclic load, went more than 6,000,000 cycles before failure. Interestingly enough, a residual stress (preload) of 108,000 psi works out to sixty percent of the UTS of the bolt, and a cyclic load of 12,000 lb. works out to about seventy-five percent of the ultimate strength of the bolt. The stress area of a ⅜x24 UNF bolt is 0.0876 sq. in. So the ultimate strength of a 180,000 psi bolt will be (180,000 x 0.0876) 15,800 lb. And 12,000 lb. is approximately seventy-five percent of 15,800 lb.

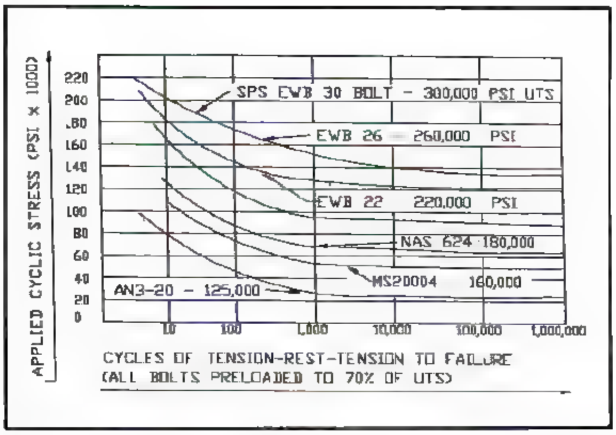
In our earlier discussion of strength of materials we discussed a rule of thumb that states that the yield strength of most steels is about sixty percent of the UTS for the low-carbon steels, up to

Recommended tightening torque for AN-3 through AN-20 and AN-73 through AN-81 bolts

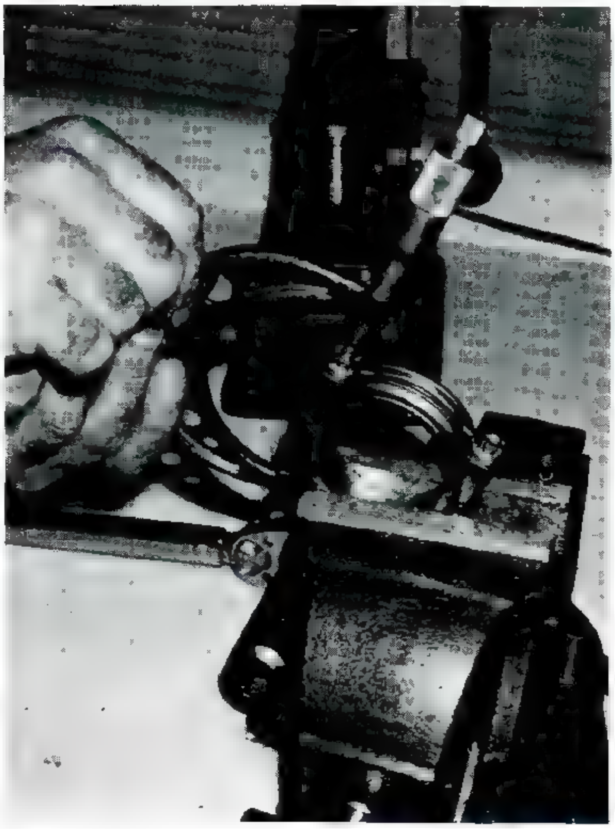
UNF Threads

Tap size	AN-365 & AN-310 Tension nuts	AN-362 & AN-320 Shear nuts
#10-32	12-15 in/lb	7-9 in/lb
¼-28	50-70 in/lb	30-40 in/lb
5/16-24	100-140 in/lb	60-85 in/lb
⅜-24	160-190 in/lb	95-110 in/lb
7/16-20	450-500 in/lb	270-300 in/lb
½-20	480-690 in/lb	290-410 in/lb
9/16-18	800-1,000 in/lb	480-600 in/lb
⅝-18	1,100-1,300 in/lb	600-780 in/lb
UNC Threads		
#10-24	20-25 in/lb	12-15 in/lb
¼-20	40-50 in/lb	25-30 in/lb
5/16-18	80-90 in/lb	48-55 in/lb
⅜-16	160-185 in/lb	95-100 in/lb
7/16-14	235-255 in/lb	140-155 in/lb
½-20	400-480 in/lb	240-290 in/lb
9/16-12	500-700 in/lb	300-420 in/lb
⅝-11	700-900 in/lb	420-540 in/lb

Recommended tightening torque sequence for AN-3 through AN-20 airframe bolts and AN-73 through AN-81 engine bolts. All values are for clean and dry cadmium-plated nuts and bolts without a lubricant. Use of a thread lubricant will make these values invalid.



The effect of upgrading bolt strength on fatigue life.

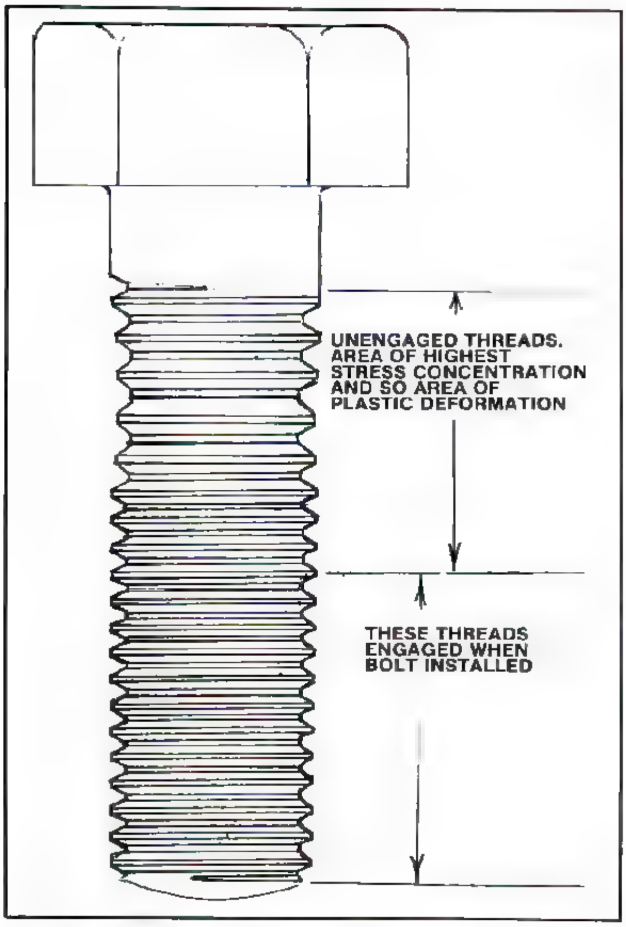


Measuring the stretch of a connecting rod bolt.

about eighty percent for the high-alloy steels. The yield strength of a typical alloy steel, heat treated to 180,000 psi UTS, is 160,000 psi or seventy-five percent of the UTS. None of this is coincidental. It *does* pay to properly tighten bolts!

Plastic deformation and the flotation test

Theory be damned! It is easy to overdue this tightening bit. Theory tells us that, due to the resistance to tightening caused by thread friction, the highest total stress that a bolt installed in tension will ever be subjected to occurs during the act of tightening—when the bolt is stressed in both tension and shear. Therefore, if the bolt doesn't fail while being tightened, when, within the parameters of its assembled endurance limit, it never should.



Stretched tension bolt. Stress has exceeded elastic limit (yield strength) of bolt, resulting in deformation of unengaged thread. The bolt is junk.

This is all very well—in theory. But, if Super Mechanic exceeds the elastic limit of a bolt while tightening it with his 18 in. wrench, then the bolt will undergo plastic deformation. It will do so locally, beginning at the root of the starting thread and progressing through the bolt section with the highest unit stress—the unengaged threads. Once this plastic deformation has occurred, the bolt will never return to its original length, or to its original strength. Even so, the theorists proclaim, the installed stress is maintained. It is this installed stress that both maintains thread tightness and determines joint strength.

You read a lot of this sort of thing in fastener manuals. It is perfectly true, so long as the elastic limit of the bolt has not been exceeded in installation and so long as the assembly is rigid. There are very few rigid assemblies and you should never preload a bolt (or a stud) quite to its yield strength. When you feel a bolt yield while being tightened, take it out and throw it away—don't even look at it. Every time you remove *any* bolt, look at the threads for signs of elongation.

If there is any sign at all or if the threads are damaged in any way, give the bolt a flotation test. This was a popular test during my days in the US Navy. Throw the metallic item under question into the nearest large body of water; if it floats, save it.

I have recently read, in a respected racing magazine, an article by an engineer with impeccable credentials who states that one should inspect bolt threads with a thread pitch gauge and, if they are out of spec, one should run a die over them before assembly! Wow! If the threads do not check out with a pitch gauge, it is because they have been stretched and the bolt is junk. Remember what I said about not trusting the experts . . . and about thread dies.

That is probably as much background as anyone other than a fastener engineer needs to know about threads in general. For that reason and because I am tired of the subject I will end this chapter and get started on the practicalities of the subject.

4

Bolts and bolted joints

Bolt Talk One: Bolt quality

Almost twenty years ago I wrote, "There is more misinformation in circulation about nuts and bolts—particularly bolts—than all of the other parts of the racing car combined. Claims and counter claims of 'equal to SAE grade 8' and 'superior to grade 8' stare smoothly from the printed page to seduce the unwary." Well, the more things change, the more they stay the same, and the statement is more true today than it was then. At the time that I wrote that particular article there was only one company hyping bad bolts to the racers as superior to SAE Grade 8, and, more important, all of the SAE grade marked hardware was being produced in this country. Now there are several corporations hyping junk, and there are inferior bolts coming from the smaller "offshore island" with head markings identical to those designated by the SAE for Grade 5 and Grade 8.

They are also counterfeiting various grades of metric bolts. The SAE is up in arms, the popular press is onto the scam with front-page stories of wheels falling off military vehicles and so on, and the FAA, realizing that many aerospace bolts are no more difficult to counterfeit than SAE bolts, is terrified. And it's not just a few ignorant purchasing agents or corrupt suppliers. At this moment, the Nuclear Regulatory Commission is investigating fifty-six companies suspected of dealing in counterfeit parts. The NRC has reported that half of the 110 nuclear plants in the United States contain counterfeit or defective fasteners. Here are some examples:

Item: Boeing discovered 2,000 counterfeit bearings in 737, 747, 757 and 767 jets manufactured between April 1986 and January 1988. Routine tests revealed that the bearings were defective. The bearings carried the markings of a US manufacturer but were actually made in Japan to less stringent standards.

Item: The FAA lists sixty-seven aircraft incidents caused by fastener failure between 1984 and 1987. The FAA is now investigating how many of the failed fasteners were counterfeit.

Item: NASA had to disassemble the Astro I space lab to remove counterfeit and defective fasteners. The cost was over $1 million. NASA had purchased the fasteners from a California distributor that was low bidder. It turned out to be one man operating out of his garage.

Item: NASA removed counterfeit bolts from the space shuttle *Discovery* before its successful launch.

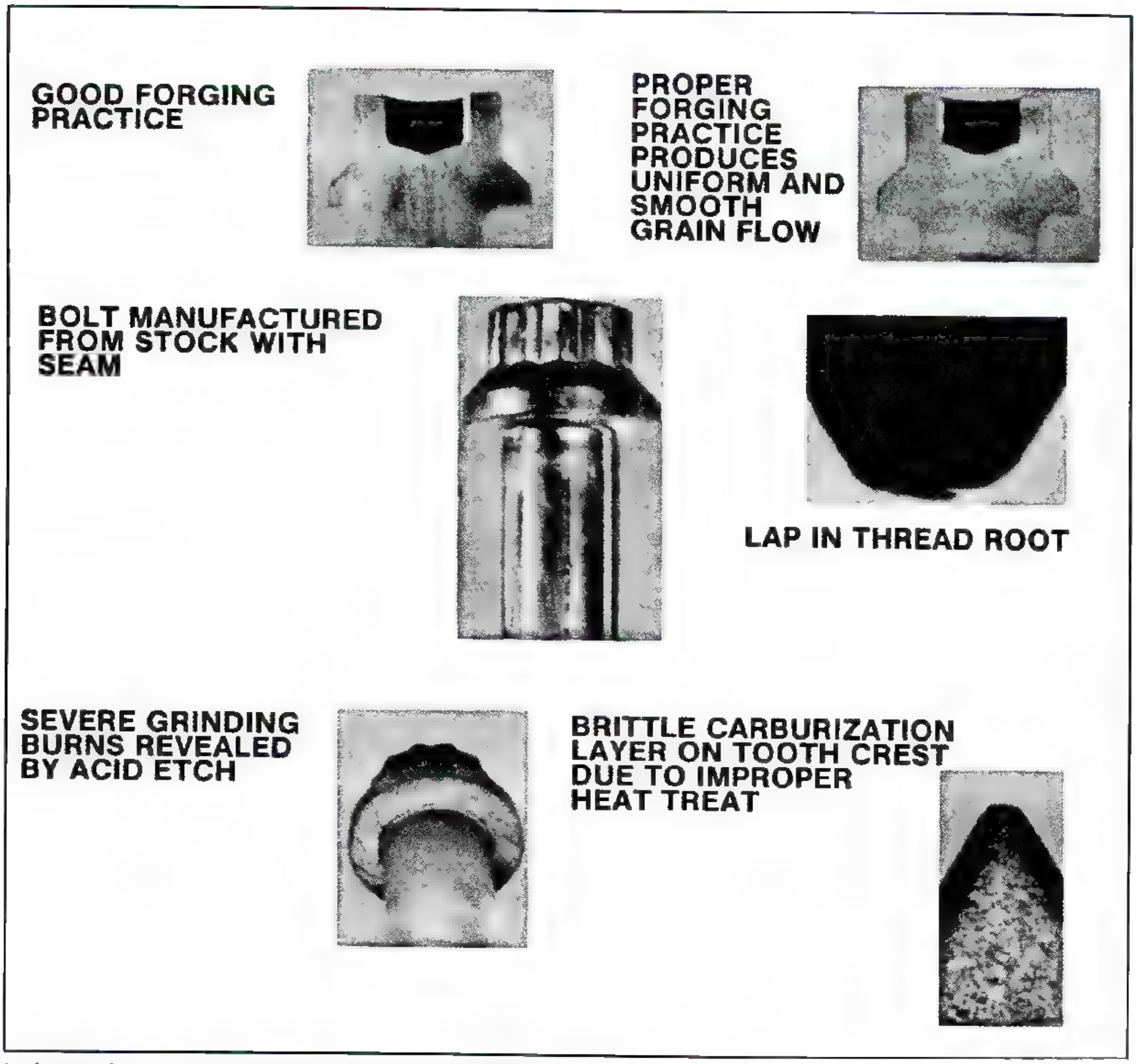
Some of these counterfeits are foreign products that are manufactured to less stringent standards and mismarked. Others are used parts that are cleaned, replated and repackaged. All of them are fraudulent; any of them can kill any of us. I said earlier that the FAA is terrified. So am I—too few purchasing agents have any knowledge of or interest in engineering, and low bid is the only criteria for altogether too much of our industrial and military purchasing.

Don't be deceived! Use no bolts that are not AN/NAS/MS certified, US-made, SAE-graded, or manufactured with UNR threads. The super-whatever bolts are not designed for fatigue-prone applications. The quality control exercised in their manufacture is minimal or nonexistent. If the parts were designed around proper bolts to begin with, you are liable to experience a fastener failure caused by metal fatigue. To an extent the same can be said about the SAE-graded bolts. The specifications to which they are manufactured are excellent. The enforcement of the specs is not, however, so you can never quite be sure of what you are getting. Further, SAE bolts are designed for tension applications and their shanks are a few thousandths undersize for shear applications.

Once you have paid for the SAE bolt, your work begins. First, you will probably have to saw off a bunch of thread. Next, you get to finish the sawed off end. Finally, you may drill a safety wire hole in the head. The FAA does not allow the use of SAE-graded bolts on aircraft and that should be some sort of clue. In fact, a recent magazine article quoted the chief engineer of one of our larger manufacturers (not identified) of SAE-graded threaded fastener as stating that he would not knowingly fly in an aircraft that used SAE-graded bolts!

The superior to SAE Grade 8 bolt

There are at least three corporations presently hyping their version of superior to SAE Grade 8 bolts. They are junk. They have always been junk and, presumably, they will always be junk. In these

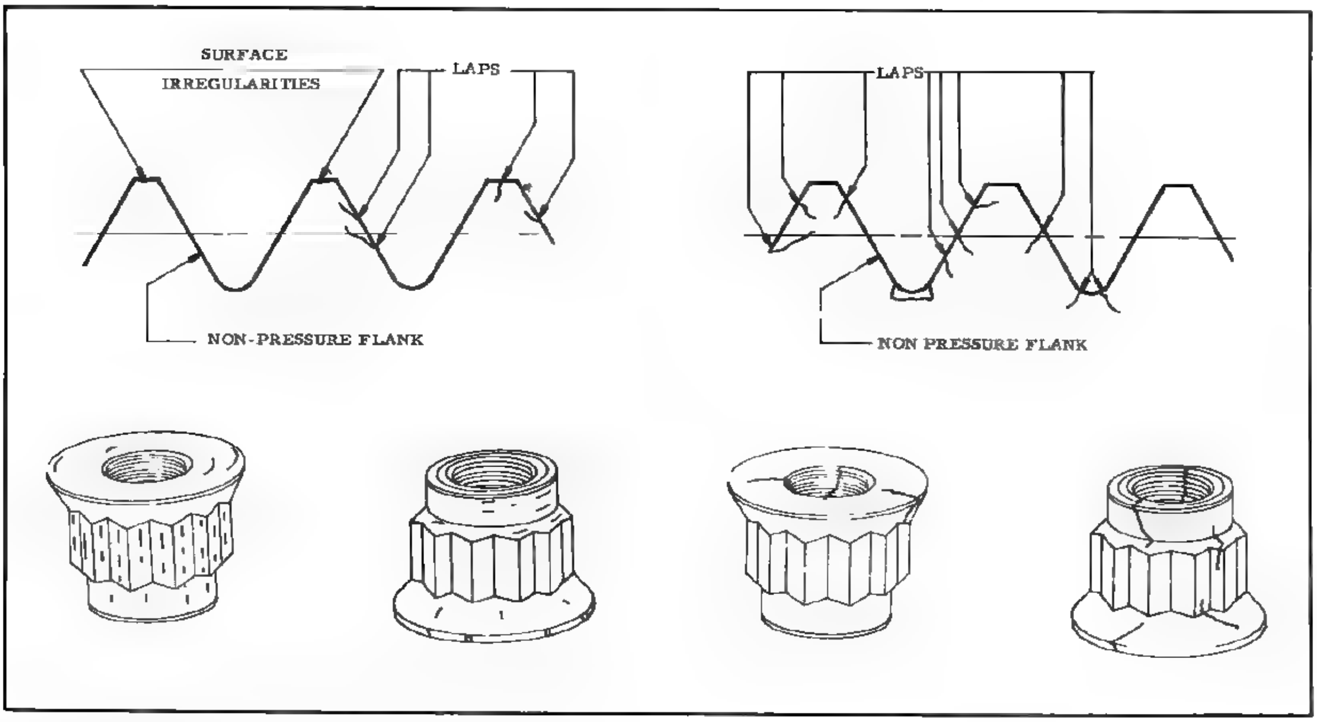


Defects in the manufacture of bolts, which are not visible to the unaided eye.

cases, superior to Grade 8 actually means they may be harder than SAE Grade 8. This usually means hard as in glass and hard as in brittle. If the SAE felt any need for a series of bolt specifications superior to their Grade 8, they would come out with Grade 9 specs, which would include toughness and fatigue resistance as well as hardness and strength. This was actually being considered a few years ago but, so far as I know, nothing came of it. If the SAE had some way of enforcing their current specifications, I would be happier and so would they.

The super-whatever claims are pure moonglow—a sales gimmick designed to rape the innocent, the unwary and the ignorant. If you use this stuff on race cars, sooner or later you are going to have a fastener failure, due either to poor quality control or poor resistance to fatigue. Seldom does a month go by without someone who should know better coming up to me with a snapped (not bent) super-whatever bolt to ask me what I think. I usually tell the individual that I think he should have his brain tested.

This is my fifth attempt to convince the racers of the world that they should not use that sort of trash. Each time, I have tried to come up with some new demonstration of the validity of my point of



Structural defects in the manufacture of nuts and bolts. At left, permissible defects; at right, non-permissible defects.

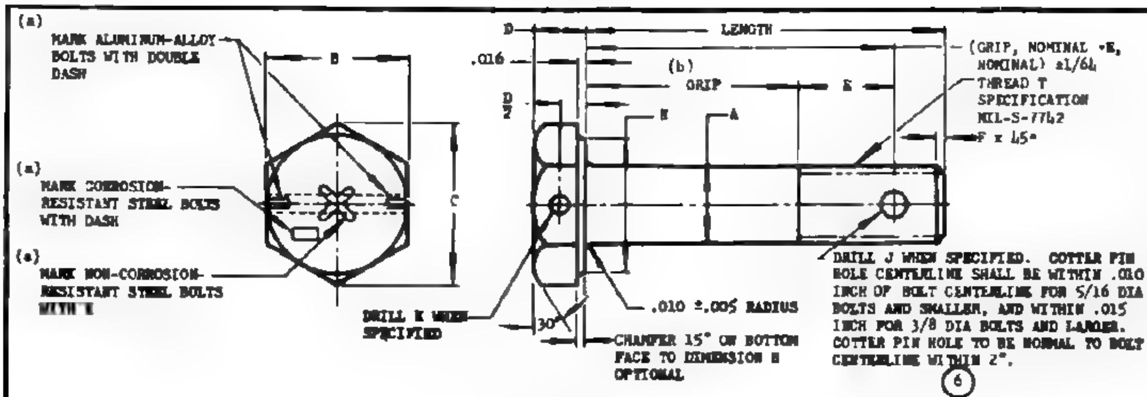
view. I have pointed out that you could take a fistful of supposedly identical super-whatever bolts to a hardness testing machine and come up with a sequence of Rockwell or Brinnel numbers from medium-soft to glass-hard. Lacking access to a hardness testing machine, make a bolt test fixture. A satisfactory one can be made from a steel block, thickness chosen to suit the bolts to be tested, and a torque wrench. Drill a series of properly sized holes in the block. Insert the test bolts through the block with a washer under each head and one between the block and each nut. The bolts are then torqued to destruction while holding the bolt head and turning the nut. If you test a fistful of super-whatever bolts, you will get a lot of different numbers, some of which will be below the manufacturer's recommended tightening torque. Next, repeat the test with aircraft bolts and see what happens.

I have also published some horror stories and some pictures to go with them. Still, I have apparently failed to convince a great many people on this subject. This time I will use more graphic examples of what can go wrong in the manufacture of bolts. A collection of examples of manufacturing defects are shown here. Some of them are not visible to the naked eye. With aerospace fasteners and with the industrial grade fasteners produced and marketed under their own name by the reputable high-strength fastener people like SPS, Allen, and Holo-Krome, you can be pretty damned sure that the manufacturer's inspection and quality control programs are going to catch these foul-ups. With the vast majority of commercial fasteners you have no such assurance. With super-whatever hype fasteners you might suspect that, if by some chance the faults should be detected, they would be ignored.

I will also point out that none of these moonglow manufacturers advertise in the real automotive trade publications like *Automotive Engineering* and *Assembly Technology Buyer's Guide.* Since those publications are where the mass market is, there has to be a reason for the nonappearance of the moonglow merchandisers. They don't even seem to advertise in the hard-core racing periodicals anymore—I guess that the racing pros have caught on.

With regard to the worthies who sell the super-whatever hardware, remember that one of Murphy's laws states, "All claims made by a manufacturer's sales representative should be multiplied by a factor of 0.25." These people are salesmen, not engineers. Their ignorance of the realities of fastening is awesome.

The economics of the situation are a bit weird. You will often be asked to pay more for a super-whatever bolt than you will for the same size bolt in SAE Grade 8 or for the equivalent aircraft certified bolt. It is all very strange—I guess someone has to pay for the hype.



BASIC AN PART NO.	THREAD T	A DIA		B		C REF	D		E REF	F		(c) H DIA	J DRILL DIA +.010 -.000	K DRILL DIA +.010 -.000
		MAX	MIN	MAX	MIN		MAX	MIN		MAX	MIN	MIN		
AN3	NO. 10-32 NF-3A	.189	.186	.377	.365	.430	.141	.109	17/64	.047	.015	.359	.070	.046
AN4	1/4 -28 UNF-3A	.249	.246	.440	.428	.510	.172	.140	5/16	.047	.015	.422	.076	.046
AN5	5/16-24 UNF 3A	.312	.309	.502	.490	.580	.204	.172	23/64	.063	.031	.484	.076	.070
AN6	3/8 -24 UNF-3A	.374	.371	.565	.553	.650	.235	.203	7/16	.063	.031	.547	.106	.070
AN7	7/16-20 UNF-3A	.437	.433	.627	.615	.720	.266	.234	31/64	.063	.031	.609	.106	.070
AN8	1/2 -20 UNF-3A	.499	.495	.752	.740	.870	.297	.265	39/64	.063	.031	.734	.106	.070
AN9	9/16-18 UNF-3A	.562	.558	.877	.865	1.010	.328	.296	21/32	.078	.046	.859	.141	.070
AN10	5/8 -18 UNF-3A	.624	.620	.940	.928	1.090	.360	.328	47/64	.078	.046	.922	.141	.070
AN12	3/4 -16 UNF-3A	.749	.744	1.066	1.053	1.230	.422	.390	7/8	.078	.046	1.047	.141	.070
AN14	7/8 -14 UNF-3A	.874	.869	1.253	1.240	1.440	.485	.453	63/64	.094	.062	1.234	.141	.070
AN16	1 -14 NF-3A	.999	.993	1.441	1.428	1.660	.547	.515	1-3/32	.094	.062	1.422	.141	.070
AN18	1-1/8 -12 UNF-3A	1.124	1.118	1.628	1.615	1.880	.610	.578	1-3/16	.110	.078	1.609	.141	.070
AN20	1-1/4 -12 UNF-3A	1.249	1.243	1.815	1.802	2.090	.672	.640	1-3/8	.110	.078	1.796	.141	.070

6 SEE SHEET 3 FOR NOTES (a) AND (b).

6 (c) THE DIAMETER OF THE WASHER FACE SHALL NOT EXCEED THE ACTUAL WIDTH ACROSS FLATS.

MATERIAL: NON-CORROSION-RESISTANT STEEL, CORROSION-RESISTANT STEEL OR ALUMINUM ALLOY. SEE PROCUREMENT SPECIFICATION.

FINISH: SEE PROCUREMENT SPECIFICATION.

ADD C BEFORE DASH NUMBER FOR CORROSION-RESISTANT STEEL BOLT.
ADD DD BEFORE DASH NUMBER FOR ALUMINUM-ALLOY BOLT.
ADD A AFTER DASH NUMBER FOR UNDRILLED BOLT. SEE ILLUSTRATION.
ADD H BEFORE DASH NUMBER FOR BOLT WITH DRILLED HEAD AND SHANK. SEE ILLUSTRATION.
ADD H BEFORE DASH NUMBER AND A AFTER DASH NUMBER FOR BOLT WITH DRILLED HEAD ONLY. SEE ILLUSTRATION.

EXAMPLES OF PART NUMBERS:

AN6-10 = 3/8 NON-CORROSION-RESISTANT STEEL BOLT 1-5/64 LONG, 7/16 GRIP WITH DRILLED SHANK ONLY. SEE ILLUSTRATION.
AN6C10 = 3/8 CORROSION-RESISTANT STEEL BOLT 1-5/64 LONG, 7/16 GRIP WITH DRILLED SHANK ONLY. SEE ILLUSTRATION.
AN6DD10 = 3/8 ALUMINUM-ALLOY BOLT 1-5/64 LONG, 7/16 GRIP WITH DRILLED SHANK ONLY. SEE ILLUSTRATION.
AN6DD10A = 3/8 ALUMINUM-ALLOY BOLT 1-5/64 LONG, 7/16 GRIP, UNDRILLED SHANK AND HEAD. SEE ILLUSTRATION.
AN6DDH10 = 3/8 ALUMINUM-ALLOY BOLT 1-5/64 LONG, 7/16 GRIP WITH DRILLED HEAD AND SHANK. SEE ILLUSTRATION.
AN6DDH10A = 3/8 ALUMINUM-ALLOY BOLT 1-5/64 LONG, 7/16 GRIP WITH DRILLED HEAD ONLY. SEE ILLUSTRATION.

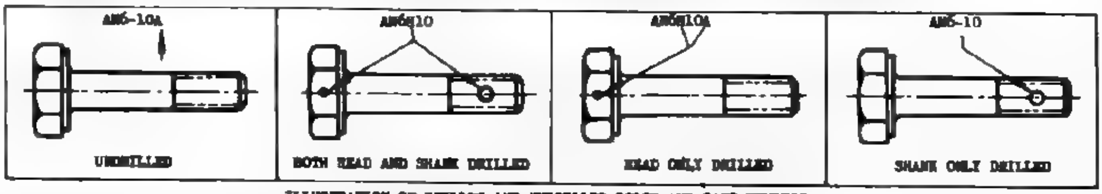


ILLUSTRATION OF DRILLED AND UNDRILLED BOLTS AND PART NUMBERS

6 BOLTS SHALL BE FREE FROM ALL HANGING BURRS AND SLIVERS WHICH MIGHT BECOME DISLODGED UNDER USAGE. COUNTERSINKING OF DRILLED HOLES IN HEAD IS MANDATORY. COUNTERSINKING OF DRILLED HOLES IN SHANK IS OPTIONAL.

DIMENSIONS IN INCHES. UNLESS OTHERWISE SPECIFIED, TOLERANCES: DECIMALS ±.010, ANGLES ±5°.

APPROVED 10 May 43 REVISED 4 20 Jan 49 5 24 Mar 52 6 15 Mar 55

PROCUREMENT SPECIFICATION	AIR FORCE-NAVY AERONAUTICAL STANDARD	AN3 THRU AN20
MIL-B-6812	BOLT - MACHINE, AIRCRAFT	SHEET 1 OF 4

Air Force-Navy specifications for the AN-3 through AN-20 airframe bolt.

DASH NO.	AN3 GRIP ±1/64	AN3 LENGTH +1/32 -1/64	AN4 GRIP ±1/64	AN4 LENGTH +1/32 -1/64	AN5 GRIP ±1/64	AN5 LENGTH +1/32 -1/64	AN6 GRIP ±1/64	AN6 LENGTH +1/32 -1/64	AN7 GRIP ±1/64	AN7 LENGTH +1/32 -1/64	AN8 GRIP ±1/64	AN8 LENGTH +1/32 -1/64	AN9 GRIP ±1/64	AN9 LENGTH +1/32 -1/64
3	1/16	15/32	1/16	15/32										
4	1/8	17/32	1/16	17/32	1/16	19/32								
5	1/4	21/32	3/16	21/32	3/16	23/32	1/16	45/64	1/16	23/32				
6	3/8	25/32	5/16	25/32	5/16	27/32	3/16	53/64	3/16	27/32	1/16	27/32	1/16	31/32
7	1/2	29/32	7/16	29/32	7/16	31/32	5/16	61/64	5/16	31/32	3/16	31/32	1/8	1- 1/32
10	5/8	1- 1/32	9/16	1- 1/32	9/16	1- 3/32	7/16	1- 5/64	7/16	1- 3/32	5/16	1- 3/32	1/4	1- 5/32
11	3/4	1- 5/32	11/16	1- 5/32	11/16	1- 7/32	9/16	1-13/64	9/16	1- 7/32	7/16	1- 7/32	3/8	1- 9/32
12	7/8	1- 9/32	13/16	1- 9/32	13/16	1-11/32	11/16	1-21/64	11/16	1-11/32	9/16	1-11/32	1/2	1-13/32
13	1	1-13/32	15/16	1-13/32	15/16	1-15/32	13/16	1-29/64	13/16	1-15/32	11/16	1-15/32	5/8	1-17/32
14	1-1/8	1-17/32	1- 1/16	1-17/32	1- 1/16	1-19/32	15/16	1-37/64	15/16	1-19/32	13/16	1-19/32	3/4	1-21/32
15	1-1/4	1-21/32	1- 3/16	1-21/32	1- 3/16	1-23/32	1- 1/16	1-45/64	1- 1/16	1-23/32	15/16	1-23/32	7/8	1-25/32
16	1-3/8	1-25/32	1- 5/16	1-25/32	1- 5/16	1-27/32	1- 3/16	1-53/64	1- 3/16	1-27/32	1- 1/16	1-27/32	1	1-29/32
17	1-1/2	1-29/32	1- 7/16	1-29/32	1- 7/16	1-31/32	1- 5/16	1-61/64	1- 5/16	1-31/32	1- 3/16	1-31/32	1-1/8	2- 1/32
20	1-5/8	2- 1/32	1- 9/16	2- 1/32	1- 9/16	2- 3/32	1- 7/16	2- 5/64	1- 7/16	2- 3/32	1- 5/16	2- 3/32	1-1/4	2- 5/32
21	1-3/4	2- 5/32	1-11/16	2- 5/32	1-11/16	2- 7/32	1- 9/16	2-13/64	1- 9/16	2- 7/32	1- 7/16	2- 7/32	1-3/8	2- 9/32
22	1-7/8	2- 9/32	1-13/16	2- 9/32	1-13/16	2-11/32	1-11/16	2-21/64	1-11/16	2-11/32	1- 9/16	2-11/32	1-1/2	2-13/32
23	2	2-13/32	1-15/16	2-13/32	1-15/16	2-15/32	1-13/16	2-29/64	1-13/16	2-15/32	1-11/16	2-15/32	1-5/8	2-17/32
24	2-1/8	2-17/32	2- 1/16	2-17/32	2- 1/16	2-19/32	1-15/16	2-37/64	1-15/16	2-19/32	1-13/16	2-19/32	1-3/4	2-21/32
25	2-1/4	2-21/32	2- 3/16	2-21/32	2- 3/16	2-23/32	2- 1/16	2-45/64	2- 1/16	2-23/32	1-15/16	2-23/32	1-7/8	2-25/32
26	2-3/8	2-25/32	2- 5/16	2-25/32	2- 5/16	2-27/32	2- 3/16	2-53/64	2- 3/16	2-27/32	2- 1/16	2-27/32	2	2-29/32
27	2-1/2	2-29/32	2- 7/16	2-29/32	2- 7/16	2-31/32	2- 5/16	2-61/64	2- 5/16	2-31/32	2- 3/16	2-31/32	2-1/8	3- 1/32
30	2-5/8	3- 1/32	2- 9/16	3- 1/32	2- 9/16	3- 3/32	2- 7/16	3- 5/64	2- 7/16	3- 3/32	2- 5/16	3- 3/32	2-1/4	3- 5/32
31	2-3/4	3- 5/32	2-11/16	3- 5/32	2-11/16	3- 7/32	2- 9/16	3-13/64	2- 9/16	3- 7/32	2- 7/16	3- 7/32	2-3/8	3- 9/32
32	2-7/8	3- 9/32	2-13/16	3- 9/32	2-13/16	3-11/32	2-11/16	3-21/64	2-11/16	3-11/32	2- 9/16	3-11/32	2-1/2	3-13/32
33	3	3-13/32	2-15/16	3-13/32	2-15/16	3-15/32	2-13/16	3-29/64	2-13/16	3-15/32	2-11/16	3-15/32	2-5/8	3-17/32
34	3-1/8	3-17/32	3- 1/16	3-17/32	3- 1/16	3-19/32	2-15/16	3-37/64	2-15/16	3-19/32	2-13/16	3-19/32	2-3/4	3-21/32
35	3-1/4	3-21/32	3- 3/16	3-21/32	3- 3/16	3-23/32	3- 1/16	3-45/64	3- 1/16	3-23/32	2-15/16	3-23/32	2-7/8	3-25/32
36	3-3/8	3-25/32	3- 5/16	3-25/32	3- 5/16	3-27/32	3- 3/16	3-53/64	3- 3/16	3-27/32	3- 1/16	3-27/32	3	3-29/32
37	3-1/2	3-29/32	3- 7/16	3-29/32	3- 7/16	3-31/32	3- 5/16	3-61/64	3- 5/16	3-31/32	3- 3/16	3-31/32	3-1/8	4- 1/32
40	3-5/8	4- 1/32	3- 9/16	4- 1/32	3- 9/16	4- 3/32	3- 7/16	4- 5/64	3- 7/16	4- 3/32	3- 5/16	4- 3/32	3-1/4	4- 5/32
41	3-3/4	4- 5/32	3-11/16	4- 5/32	3-11/16	4- 7/32	3- 9/16	4-13/64	3- 9/16	4- 7/32	3- 7/16	4- 7/32	3-3/8	4- 9/32
42	3-7/8	4- 9/32	3-13/16	4- 9/32	3-13/16	4-11/32	3-11/16	4-21/64	3-11/16	4-11/32	3- 9/16	4-11/32	3-1/2	4-13/32
43	4	4-13/32	3-15/16	4-13/32	3-15/16	4-15/32	3-13/16	4-29/64	3-13/16	4-15/32	3-11/16	4-15/32	3-5/8	4-17/32
44	4-1/8	4-17/32	4- 1/16	4-17/32	4- 1/16	4-19/32	3-15/16	4-37/64	3-15/16	4-19/32	3-13/16	4-19/32	3-3/4	4-21/32
45	4-1/4	4-21/32	4- 3/16	4-21/32	4- 3/16	4-23/32	4- 1/16	4-45/64	4- 1/16	4-23/32	3-15/16	4-23/32	3-7/8	4-25/32
46	4-3/8	4-25/32	4- 5/16	4-25/32	4- 5/16	4-27/32	4- 3/16	4-53/64	4- 3/16	4-27/32	4- 1/16	4-27/32	4	4-29/32
47	4-1/2	4-29/32	4- 7/16	4-29/32	4- 7/16	4-31/32	4- 5/16	4-61/64	4- 5/16	4-31/32	4- 3/16	4-31/32	4-1/8	5- 1/32
50	4-5/8	5- 1/32	4- 9/16	5- 1/32	4- 9/16	5- 3/32	4- 7/16	5- 5/64	4- 7/16	5- 3/32	4- 5/16	5- 3/32	4-1/4	5- 5/32
51	4-3/4	5- 5/32	4-11/16	5- 5/32	4-11/16	5- 7/32	4- 9/16	5-13/64	4- 9/16	5- 7/32	4- 7/16	5- 7/32	4-3/8	5- 9/32
52	4-7/8	5- 9/32	4-13/16	5- 9/32	4-13/16	5-11/32	4-11/16	5-21/64	4-11/16	5-11/32	4- 9/16	5-11/32	4-1/2	5-13/32
53	5	5-13/32	4-15/16	5-13/32	4-15/16	5-15/32	4-13/16	5-29/64	4-13/16	5-15/32	4-11/16	5-15/32	4-5/8	5-17/32
54	5-1/8	5-17/32	5- 1/16	5-17/32	5- 1/16	5-19/32	4-15/16	5-37/64	4-15/16	5-19/32	4-13/16	5-19/32	4-3/4	5-21/32
55	5-1/4	5-21/32	5- 3/16	5-21/32	5- 3/16	5-23/32	5- 1/16	5-45/64	5- 1/16	5-23/32	4-15/16	5-23/32	4-7/8	5-25/32
56	5-3/8	5-25/32	5- 5/16	5-25/32	5- 5/16	5-27/32	5- 3/16	5-53/64	5- 3/16	5-27/32	5- 1/16	5-27/32	5	5-29/32
57	5-1/2	5-29/32	5- 7/16	5-29/32	5- 7/16	5-31/32	5- 5/16	5-61/64	5- 5/16	5-31/32	5- 3/16	5-31/32	5-1/8	6- 1/32
60	5-5/8	6- 1/32	5- 9/16	6- 1/32	5- 9/16	6- 3/32	5- 7/16	6- 5/64	5- 7/16	6- 3/32	5- 5/16	6- 3/32	5-1/4	6- 5/32
61	5-3/4	6- 5/32	5-11/16	6- 5/32	5-11/16	6- 7/32	5- 9/16	6-13/64	5- 9/16	6- 7/32	5- 7/16	6- 7/32	5-3/8	6- 9/32
62	5-7/8	6- 9/32	5-13/16	6- 9/32	5-13/16	6-11/32	5-11/16	6-21/64	5-11/16	6-11/32	5- 9/16	6-11/32	5-1/2	6-13/32
63	6	6-13/32	5-15/16	6-13/32	5-15/16	6-15/32	5-13/16	6-29/64	5-13/16	6-15/32	5-11/16	6-15/32	5-5/8	6-17/32
64	6-1/8	6-17/32	6- 1/16	6-17/32	6- 1/16	6-19/32	5-15/16	6-37/64	5-15/16	6-19/32	5-13/16	6-19/32	5-3/4	6-21/32
65	6-1/4	6-21/32	6- 3/16	6-21/32	6- 3/16	6-23/32	6- 1/16	6-45/64	6- 1/16	6-23/32	5-15/16	6-23/32	5-7/8	6-25/32
66	6-3/8	6-25/32	6- 5/16	6-25/32	6- 5/16	6-27/32	6- 3/16	6-53/64	6- 3/16	6-27/32	6- 1/16	6-27/32	6	6-29/32
67	6-1/2	6-29/32	6- 7/16	6-29/32	6- 7/16	6-31/32	6- 5/16	6-61/64	6- 5/16	6-31/32	6- 3/16	6-31/32	6-1/8	7- 1/32
70	6-5/8	7- 1/32	6- 9/16	7- 1/32	6- 9/16	7- 3/32	6- 7/16	7- 5/64	6- 7/16	7- 3/32	6- 5/16	7- 3/[illegible]	6-1/4	7- 5/32
71	6-3/4	7- 5/32	6-11/16	7- 5/32	6-11/16	7 7/32	6- 9/16	7-13/64	6- 9/16	7- 7/32	6- 7/16	7- 7/32	6-3/8	7- 9/32
72	6-7/8	7- 9/32	6-13/16	7- 9/32	6-13/16	7-11/32	6-11/16	7-21/64	6-11/16	7-11/32	6- 9/16	7-11/32	6-1/2	7-13/32
73	7	7-13/32	6-15/16	7-13/32	6-15/16	7-15/32	6-13/16	7-29/64	6-13/16	7-15/32	6-11/16	7-15/32	6-5/8	7-17/32
74	7-1/8	7-17/32	7- 1/16	7-17/32	7- 1/16	7-19/32	6-15/16	7-37/64	6-15/16	7-19/32	6-13/16	7-19/32	6-3/4	7-21/32
75	7-1/4	7-21/32	7- 3/16	7-21/32	7- 3/16	7-23/32	7- 1/16	7-45/64	7- 1/16	7-23/32	6-15/16	7-23/32	6-7/8	7-25/32
76	7-3/8	7-25/32	7- 5/16	7-25/32	7- 5/16	7-27/32	7- 3/16	7 53/64	7- 3/16	7-27/32	7- 1/16	7-27/32	7	7-29/32
77	7-1/2	7-29/32	7- 7/16	7-29/32	7- 7/16	7-31/32	7- 5/16	7-61/64	7- 5/16	7-31/32	7- 3/16	7-31/32	7-1/8	8- 1/32
80	7-5/8	8- 1/32.	7- 9/16	8- 1/32	7- 9/16	8- 3/32	7- 7/16	8- 5/64	7- 7/16	8- 3/32	7- 5/16	8- 3/32	7-1/4	8- 5/32
81					7-11/16	8- 7/32					7- 7/16	8- 7/32		
82											7- 9/16	8-11/32		
83											7-11/16	8-15/32		
84											7-13/16	8-19/32		
85							8- 1/16	8-45/64			7-15/16	8-23/32		
86											8- 1/16	8-27/32		
87											8- 3/16	8-31/32		

APPROVED 10 May 43 REVISED (4) 20 Jan 48 (5) 24 Mar 52

(5) PROCUREMENT SPECIFICATION MIL-B-6812

AIR FORCE-NAVY AERONAUTICAL STANDARD

(5) BOLT - MACHINE, AIRCRAFT

AN3 THRU AN20

(5) SHEET 2 OF 4

DASH NO.	AN10 GRIP ±1/64	AN10 LENGTH +1/32 -1/64	AN12 GRIP ±1/64	AN12 LENGTH +1/32 -1/64	AN14 GRIP ±1/64	AN14 LENGTH +1/32 -1/64	AN16 GRIP ±1/64	AN16 LENGTH +1/32 -1/64	[illegible] GRIP ±1/64	[illegible] LENGTH +1/32 -1/64	AN18 GRIP ±1/64	AN18 LENGTH +1/32 -1/64	AN20 GRIP ±1/64	AN20 LENGTH +1/32 -1/64
7	1/16	1- 1/64												
10	3/16	1- 9/64	1/16	1- 5/32										
11	5/16	1-17/64	3/16	1- 9/32	1/16	1- 5/16								
12	7/16	1-25/64	5/16	1-13/32	3/16	1- 7/16	1/8	1-1/2	1/8	1-1/2				
13	9/16	1-33/64	7/16	1-17/32	5/16	1- 9/16	1/4	1-5/8	1/4	1-5/8	1/16	1- 9/16		
14	11/16	1-41/64	9/16	1-21/32	7/16	1-11/16	3/8	1-3/4	3/8	1-3/4	3/16	1-11/16		
15	13/16	1-49/64	11/16	1-25/32	9/16	1-13/16	1/2	1-7/8	1/2	1-7/8	5/16	1-13/16	1/8	1-13/16
16	15/16	1-57/64	13/16	1-29/32	11/16	1-15/16	5/8	2	5/8	2	7/16	1-15/16	1/4	1-15/16
17	1- 1/16	2- 1/64	15/16	2- 1/32	13/16	2- 1/16	3/4	2-1/8	3/4	2-1/8	9/16	2- 1/16	3/8	2- 1/16
20	1- 3/16	2- 9/64	1- 1/16	2- 5/32	15/16	2- 3/16	7/8	2-1/4	7/8	2-1/4	11/16	2- 3/16	1/2	2- 3/16
21	1- 5/16	2-17/64	1- 3/16	2- 9/32	1- 1/16	2- 5/16	1	2-3/8	1	2-3/8	13/16	2- 5/16	5/8	2- 5/16
22	1- 7/16	2-25/64	1- 5/16	2-13/32	1- 3/16	2- 7/16	1-1/8	2-1/2	1-1/8	2-1/2	15/16	2- 7/16	3/4	2- 7/16
23	1- 9/16	2-33/64	1- 7/16	2-17/32	1- 5/16	2- 9/16	1-1/4	2-5/8	1-1/4	2-5/8	1- 1/16	2- 9/16	7/8	2- 9/16
24	1-11/16	2-41/64	1- 9/16	2-21/32	1- 7/16	2-11/16	1-3/8	2-3/4	1-3/8	2-3/4	1- 3/16	2-11/16	1	2-11/16
25	1-13/16	2-49/64	1-11/16	2-25/32	1- 9/16	2-13/16	1-1/2	2-7/8	1-1/2	2-7/8	1- 5/16	2-13/16	1-1/8	2-13/16
26	1-15/16	2-57/64	1-13/16	2-29/32	1-11/16	2-15/16	1-5/8	3	1-5/8	3	1- 7/16	2-15/16	1-1/4	2-15/16
27	2- 1/16	3- 1/64	1-15/16	3- 1/32	1-13/16	3- 1/16	1-3/4	3-1/8	1-3/4	3-1/8	1- 9/16	3- 1/16	1-3/8	3- 1/16
30	2- 3/16	3- 9/64	2- 1/16	3- 5/32	1-15/16	3- 3/16	1-7/8	3-1/4	1-7/8	3-1/4	1-11/16	3- 3/16	1-1/2	3- 3/16
31	2- 5/16	3-17/64	2- 3/16	3- 9/32	2- 1/16	3- 5/16	2	3-3/8	2	3-3/8	1-13/16	3- 5/16	1-5/8	3- 5/16
32	2- 7/16	3-25/64	2- 5/16	3-13/32	2- 3/16	3- 7/16	2-1/8	3-1/2	2-1/8	3-1/2	1-15/16	3- 7/16	1-3/4	3- 7/16
33	2- 9/16	3-33/64	2- 7/16	3-17/32	2- 5/16	3- 9/16	2-1/4	3-5/8	2-1/4	3-5/8	2- 1/16	3- 9/16	1-7/8	3- 9/16
34	2-11/16	3-41/64	2- 9/16	3-21/32	2- 7/16	3-11/16	2-3/8	3-3/4	2-3/8	3-3/4	2- 3/16	3-11/16	2	3-11/16
35	2-13/16	3-49/64	2-11/16	3-25/32	2- 9/16	3-13/16	2-1/2	3-7/8	2-1/2	3-7/8	2- 5/16	3-13/16	2-1/8	3-13/16
36	2-15/16	3-57/64	2-13/16	3-29/32	2-11/16	3-15/16	2-5/8	4	2-5/8	4	2- 7/16	3-15/16	2-1/4	3-15/16
37	3- 1/16	4- 1/64	2-15/16	4- 1/32	2-13/16	4- 1/16	2-3/4	4-1/8	2-3/4	4-1/8	2- 9/16	4- 1/16	2-3/8	4- 1/16
40	3- 3/16	4- 9/64	3- 1/16	4- 5/32	2-15/16	4- 3/16	2-7/8	4-1/4	2-7/8	4-1/4	2-11/16	4- 3/16	2-1/2	4- 3/16
41	3- 5/16	4-17/64	3- 3/16	4- 9/32	3- 1/16	4- 5/16	3	4-3/8	3	4-3/8	2-13/16	4- 5/16	2-5/8	4- 5/16
42	3- 7/16	4-25/64	3- 5/16	4-13/32	3- 3/16	4- 7/16	3-1/8	4-1/2	3-1/8	4-1/2	2-15/16	4- 7/16	2-3/4	4- 7/16
43	3- 9/16	4-33/64	3- 7/16	4-17/32	3- 5/16	4- 9/16	3-1/4	4-5/8	3-1/4	4-5/8	3- 1/16	4- 9/16	2-7/8	4- 9/16
44	3-11/16	4-41/64	3- 9/16	4-21/32	3- 7/16	4-11/16	3-3/8	4-3/4	3-3/8	4-3/4	3- 3/16	4-11/16	3	4-11/16
45	3-13/16	4-49/64	3-11/16	4-25/32	3- 9/16	4-13/16	3-1/2	4-7/8	3-1/2	4-7/8	3- 5/16	4-13/16	3-1/8	4-13/16
46	3-15/16	4-57/64	3-13/16	4-29/32	3-11/16	4-15/16	3-5/8	5	3-5/8	5	3- 7/16	4-15/16	3-1/4	4-15/16
47	4- 1/16	5- 1/64	3-15/16	5- 1/32	3-13/16	5- 1/16	3-3/4	5-1/8	3-3/4	5-1/8	3- 9/16	5- 1/16	3-3/8	5- 1/16
50	4- 3/16	5- 9/64	4- 1/16	5- 5/32	3-15/16	5- 3/16	3-7/8	5-1/4	3-7/8	5-1/4	3-11/16	5- 3/16	3-1/2	5- 3/16
51	4- 5/16	5-17/64	4- 3/16	5- 9/32	4- 1/16	5- 5/16	4	5-3/8	4	5-3/8	3-13/16	5- 5/16	3-5/8	5- 5/16
52	4- 7/16	5-25/64	4- 5/16	5-13/32	4- 3/16	5- 7/16	4-1/8	5-1/2	4-1/8		3-15/16	5- 7/16	3-3/4	5- 7/16
53	4- 9/16	5-33/64	4- 7/16	5-17/32	4- 5/16	5- 9/16	4-1/4	5-5/8	4-1/4	5-5/8	4- 1/16	5- 9/16	3-7/8	5- 9/16
54	4-11/16	5-41/64	4- 9/16	5-21/32	4- 7/16	5-11/16	4-3/8	5-3/4	4-3/8	5-3/4	4- 3/16	5-11/16	4	5-11/16
55	4-13/16	5-49/64	4-11/16	5-25/32	4- 9/16	5-13/16	4-1/2	5-7/8	4-1/2	5-7/8	4- 5/16	5-13/16	4-1/8	5-13/16
56	4-15/16	5-57/64	4-13/16	5-29/32	4-11/16	5-15/16	4-5/8	6	4-5/8	6	4- 7/16	5-15/16	4-1/4	5-15/16
57	5- 1/16	6- 1/64	4-15/16	6- 1/32	4-13/16	6- 1/16	4-3/4	6-1/8	4-3/4	6-1/8	4- 9/16	6- 1/16	4-3/8	6- 1/16
60	5- 3/16	6- 9/64	5- 1/16	6- 5/32	4-15/16	6- 3/16	4-7/8	6-1/4	4-7/8	6-1/4	4-11/16	6- 3/16	4-1/2	6- 3/16
61	5- 5/16	6-17/64	5- 3/16	6- 9/32	5- 1/16	6- 5/16	5	6-3/8	5	6-3/8	4-13/16	6- 5/16	4-5/8	6- 5/16
62	5- 7/16	6-25/64	5- 5/16	6-13/32	5- 3/16	6- 7/16	5-1/8	6-1/2	5-1/8	6-1/2	4-15/16	6- 7/16	4-3/4	6- 7/16
63	5- 9/16	6-33/64	5- 7/16	6-17/32	5- 5/16	6- 9/16	5-1/4	6-5/8	5-1/4	6-5/8	5- 1/16	6- 9/16	4-7/8	6- 9/16
64	5-11/16	6-41/64	5- 9/16	6-21/32	5- 7/16	6-11/16	5-3/8	6-3/4	5-3/8	6-3/4	5- 3/16	6-11/16	5	6-11/16
65	5-13/16	6-49/64	5-11/16	6-25/32	5- 9/16	6-13/16	5-1/2	6-7/8	5-1/2	6-7/8	5- 5/16	6-13/16	5-1/8	6-13/16
66	5-15/16	6-57/64	5-13/16	6-29/32	5-11/16	6-15/16	5-5/8	7	5-5/8	7	5- 7/16	6-15/16	5-1/4	6-15/16
67	6- 1/16	7- 1/64	5-15/16	7- 1/32	5-13/16	7- 1/16	5-3/4	7-1/8	5-3/4	7-1/8	5- 9/16	7- 1/16	5-3/8	7- 1/16
70	6- 3/16	7- 9/64	6- 1/16	7- 5/32	5-15/16	7- 3/16	5-7/8	7-1/4	5-7/8	7-1/4	5-11/16	7- 3/16	5-1/2	7- 3/16
71	6- 5/16	7-17/64	6- 3/16	7- 9/32	6- 1/16	7- 5/16	6	7-3/8	6	7-3/8	5-13/16	7- 5/16	5-5/8	7- 5/16
72	6- 7/16	7-25/64	6- 5/16	7-13/32	6- 3/16	7- 7/16	6-1/8	7-1/2	6-1/8	7-1/2	5-15/16	7- 7/16	5-3/4	7- 7/16
73	6- 9/16	7-33/64	6- 7/16	7-17/32	6- 5/16	7- 9/16	6-1/4	7-5/8	6-1/4	7-5/8	6- 1/16	7- 9/16	5-7/8	7- 9/16
74	6-11/16	7-41/64	6- 9/16	7-21/32	6- 7/16	7-11/16	6-3/8	7-3/4	6-3/8	7-3/4	6- 3/16	7-11/16	6	7-11/16
75	6-13/16	7-49/64	6-11/16	7-25/32	6- 9/16	7-13/16	6-1/2	7-7/8	6-1/2	7-7/8	6- 5/16	7-13/16	6-1/8	7-13/16
76	6-15/16	7-57/64	6-13/16	7-29/32	6-11/16	7-15/16	6-5/8	8	6-5/8	8	6- 7/16	7-15/16	6-1/4	7-15/16
77	7- 1/16	8- 1/64	6-15/16	8- 1/32	6-13/16	8- 1/16	6-3/4	8-1/8	6-3/4	8-1/8	6- 9/16	8- 1/16	6-3/8	8- 1/16
80	7- 3/16	8- 9/64	7- 1/16	8- 5/32	6-15/16	8- 3/16	6-7/8	8-1/4	6-7/8	8-1/4	6-11/16	8- 3/16	6-1/2	8- 3/16
81							7	8-3/8	7	8-3/8	6-13/16	8- 5/16	6-5/8	8- 5/16
82							7-1/8	8-1/2	7-1/8	8-1/2				

(a) MARKING MAY BE EITHER RAISED OR DEPRESSED.
THE HEIGHT OR DEPTH OF THE MARKINGS SHALL NOT EXCEED THE FOLLOWING VALUES:
.025 INCH FOR 9/16 DIAMETER BOLTS AND SMALLER;
.030 INCH FOR 5/8 DIAMETER BOLTS AND LARGER.
HEAD MARKING MAY INCLUDE MANUFACTURER'S MARKINGS.
(b) GRIP LENGTH OF BOLTS SHALL BE MEASURED FROM THE UNDERSIDE OF THE HEAD TO THE END OF THE FULL CYLINDRICAL PORTION OF THE SHANK. COMPLETE THREADS SHALL BEGIN WITHIN TWO THREAD PITCH MAXIMUM. TWO THREAD PITCH MAXIMUM MAY CONSIST OF INCOMPLETE THREAD OR EXTRUSION ANGLE.

APPROVED 10 May 43 REVISED (4) 20 Jan 48 (5) 24 Mar 52 (9) FOR CHANGES SEE SHEET 1.

PROCUREMENT SPECIFICATION	AIR FORCE-NAVY AERONAUTICAL STANDARD	AN3 THRU AN20
MIL-B-6812	BOLT - MACHINE, AIRCRAFT	SHEET 3 OF

AN-3 through AN-20 airframe bolt specifications.

The solution: The AN system

Several decades ago the powers that be, concerned with the frequency with which aircraft parts were coming unbolted and falling out of the sky, sat down and devised a set of standards for aircraft hardware known as the Air Corps/Navy or AN specifications. With the passage of time and the growth of the federal bureaucracy, this has now become Military Specification or MS, and is augmented by National Aerospace Standard or NAS. The full set of specs looks like ten encyclopedia and is not readily available to the public. Fortunately you don't need much of the information that is contained in the full set of specs. The Van Deusen Aircraft Company's hardware catalog and Aircraft Spruce and Specialty's catalog (one of the great catalogs of all time) contain virtually everything you would need to know.

Useful and informative additions to the catalogs are the *Standard Aircraft Handbook* from Aero Publishers and *The Standard Aircraft Workers' Manual* from Fletcher Aircraft. These are available at nominal cost from any general aviation store. If you feel that you need information on the high-strength NAS superbolts, you could try to coax a copy of Standard Pressed Steel's *Bolts for the Aerospace Industry* from a local distributor.

Basically, the AN, MS and NAS hardware offers a range of fasteners designed to do a job similar to ours and which are manufactured and inspected to stringent standards. Dimensions are closely controlled; surfaces are fully finished and true; and strength and hardness are consistent and dependable. They are not only strong, they are very tough. These bolts will bend before they break. Every conceivable size and configuration is available. All of this should cost a fortune, and now that the surplus market has almost completely dried up, it sometimes seems that it does. However, the right stuff is still less expensive than the super-whatever junk. The AN and MS items are actually price-competitive with both SAE Grade 8 and the best commercial socket-head cap screws. And there is still some surplus available—at least in Los Angeles.

Bolt Talk Two: The right stuff

For our purposes we need to consider only five series of aerospace bolts. For most fine threaded applications we should use the AN-3 through AN-20 series of airframe bolts (also called six-digit AN bolts. These are hexagon-headed tension bolts available in all UNF sizes. Length of thread is consistent in each diameter (about two diameters) and is calculated to accept a standard AN washer, a standard AN full-height tension nut and leave 3½ threads exposed after the nut. Grip lengths are available from 1/16 in. to 6 in. in increments of 1/8 in., so the proper length bolt is available for just about any application.

They are manufactured in carbon steel (SAE 2330, heat treated and cadmium plated), stainless steel and in aluminum (2024–T4). Threads are to UNF Class 3A specifications. The carbon steel series is the only one that is readily available. Fortunately the steel bolts are both stronger and more fatigue

Standard AN thread lengths

Bolt diameter and thread pitch	Thread length (for full height nut)	Thread length (for shear nut)
#10–32	0.334	0.236
1/4–28	0.412	0.293
5/16–24	0.480	0.351
3/8–24	0.560	0.400
7/16–20	0.665	0.466
1/2–20	0.761	0.516

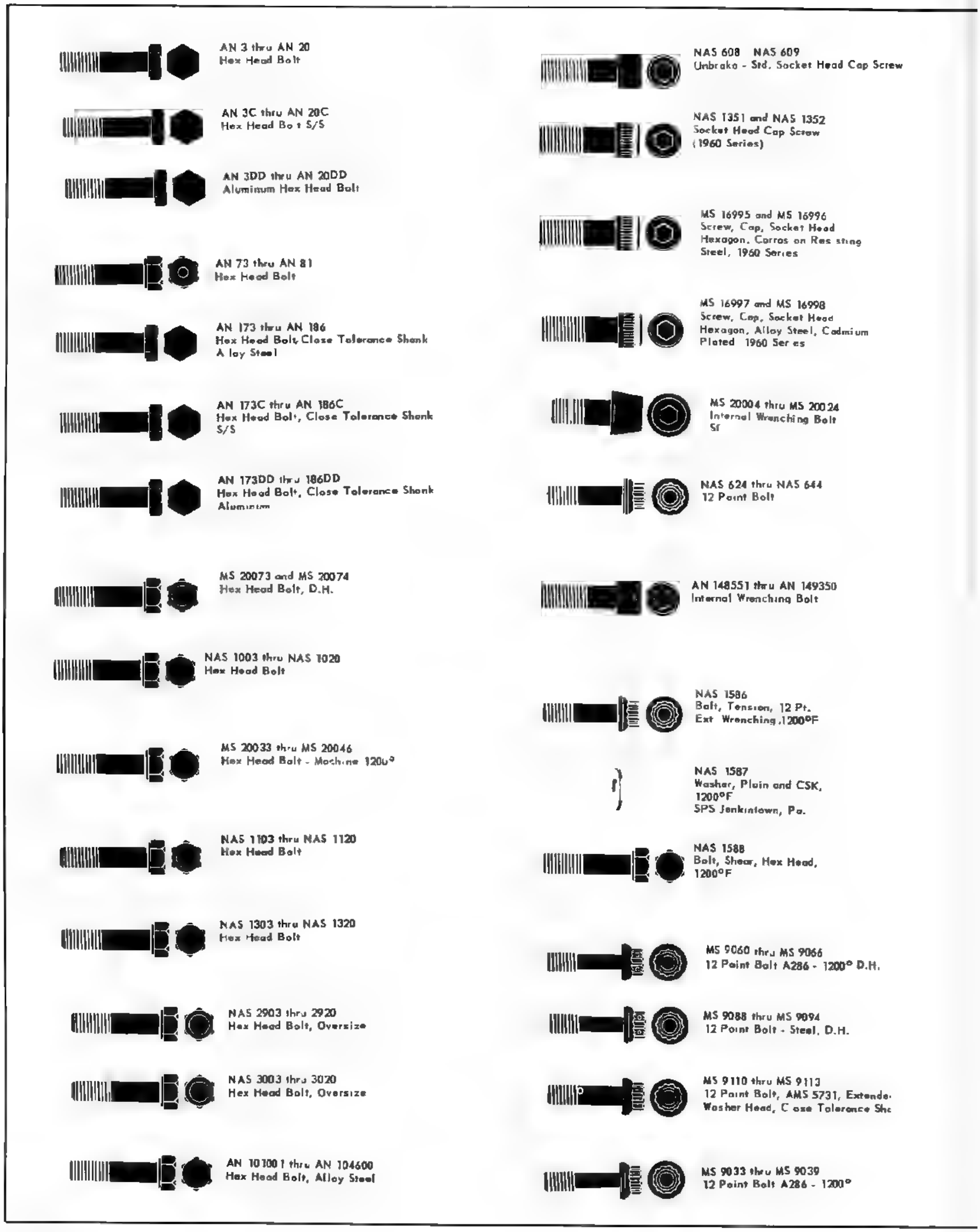
Standard AN tension and shear bolt thread lengths.

Shear and tensile strength of AN–3 through AN–20 bolts

Source: ANC–5

AN #	Diameter (in.)	Allowable single shear load (lb.) @ full diam.	Allowable tensile load (lb.) @ thread root diam.	Area of cross section (full diam.)	Stress area UNF (sq. in.)
AN–3	3/16	2,070	N/A	0.0276	0.0199
AN–4	1/4	3,680	4,080	0.0491	0.0362
AN–5	5/16	5,750	6,500	0.0767	0.0579
AN–6	3/8	8,280	10,100	0.1105	0.0876
AN–7	7/16	11,200	13,600	0.1503	0.1185
AN–8	1/2	12,760	18,500	0.1963	0.1597
AN–9	9/16	18,700	23,600	0.2485	0.2026
AN–10	5/8	23,000	30,100	0.3068	0.2555

Shear and tensile strength of AN–3 through AN–20 bolts.



Common AN, MS and NAS bolts.

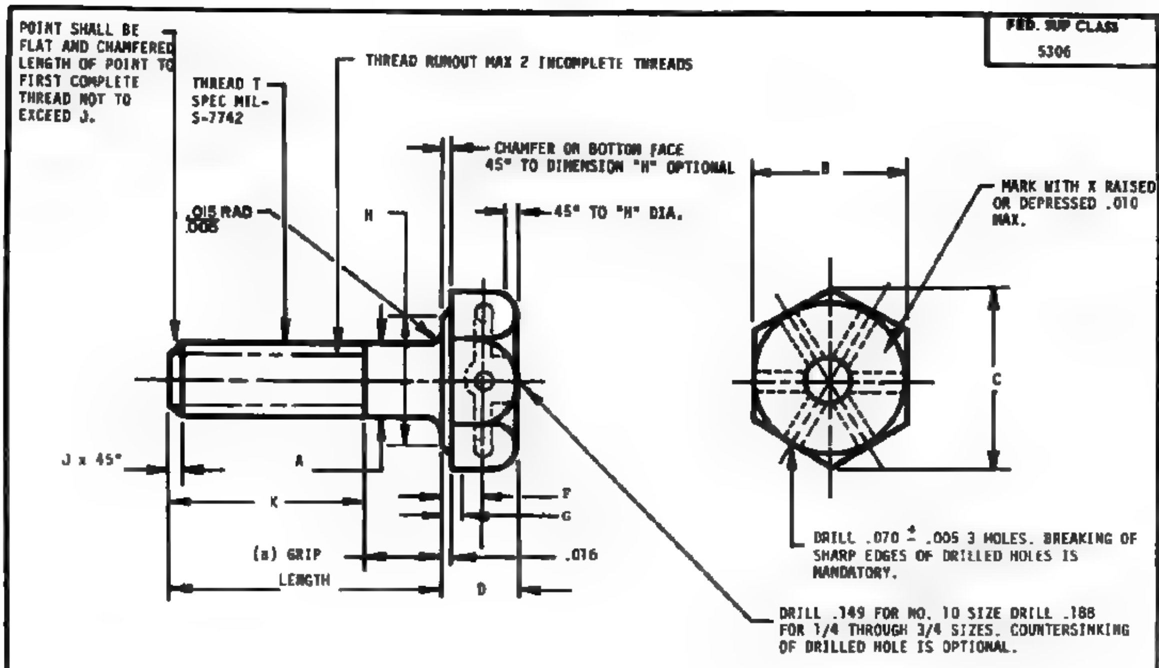
resistant than the others and so are the only series of real interest to us. Minimum ultimate tensile strength in carbon steel is 125,000 psi. Both tensile yield strength and ultimate shear strength is about 75,000 psi. This means that an AN-6 bolt (⅜x24 UNF), properly installed and tightened, has an ultimate strength in tension of 10,100 lb. and an ultimate strength in single shear of 8,280 lb. In double shear the number is 16,400 lb. If you or I ever approach that kind of load without hitting something very solid, we had better buy new drawing boards.

The AN bolts have true bearing surfaces under the heads. This may not sound like a big deal. It is. If the bearing surface of a bolt is not normal to bolt axis, then a bending load will be imposed as the bolt is tightened and premature fatigue failure is only a matter of (not much) time. By a simple alteration in the basic part number, AN bolts are available with the heads and/or shanks drilled for safety wire or cotter pins, thus saving hours of tedious labor and many broken drill bits. Given the option, I always buy my bolts with drilled heads. They don't cost any more; the hole doesn't weaken the bolt; and you never know when it is going to come in handy. A reasonable selection of AN-3 through AN-20 bolts should be available at any general aviation store.

For coarse threaded applications you should use the MS20074 series, which used to be AN-73A through AN-81A series, sometimes referred to as engine bolts. These bolts are made to the same general specifications as the AN-3 through AN-20 bolts but with slightly thicker heads and UNC threads. They are also available in UNF as MS20073, but are very difficult to find and offer no advantage over AN-3 through AN-20.

The illustration shows the strengths of the coarse threaded engine bolts. Note that the cross-sectional area of the bolt shank and the rated single shear strength is the same as the AN-3 through AN-20 bolts. Due to the UNC thread, the stress area and therefore the allowable tension strength are considerably less. While we should be aware of this reduction in tensile strength, it is no big deal—with the notable exception of cylinder head bolts and main bearing cap bolts (which should be studs anyway), most coarse threaded applications are relatively lightly loaded. This means that an SAE Grade 8 bolt or a good commercial socket-headed cap screw would be perfectly safe. As we have seen, the trouble with the commercial bolts is that we have to do too much work on them before we can use them, especially since we usually have to safety wire our coarse threaded bolts. Since the AN engine bolts were designed to hold the cylinder barrels of radial engines to their crankcases, they come cross-drilled with not one but three safety wire through holes so that you can always get the wire through.

FED. SUP CLASS
5306



SIZE DASH NO.	THREAD T	A DIA.		B		(b) C REF	D		F MIN	G +.000 -.010	H DIA.		J		RATED STRENGTH, LBS	
		MAX	MIN.	MAX.	MIN.		MAX.	MIN.			MAX.	MIN.	MAX.	MIN.	(c) TENSION AT ROOT DIA.	SINGLE SHEAR AT FULL DIA.
03-	NO. 10-24UNC-3A	.189	.186	.376	.367	.430	.203	.172	.039	.120	.391	.359	.047	.015	1 800	2 125
04-	1/4-20 UNC-3A	.249	.246	.439	.430	.510	.234	.203	.052	.144	.454	.422			3 360	3 680
05-	5/16 -18 UNC-3A	.312	.309	.501	.492	.580	.297	266	.078	.172	.516	.484	.063	.031	5 660	5 750
06-	3/8 -16 UNC-3A	.374	.371	.564	.554	.650					.579	.547			8 470	8 800
07-	7/16 -14 UNC-3A	.437	.433	.689	.679	.790	.344	.313			.704	.672			11 680	11 250
08-	1/2-13 UNC-3A	.499	.495	.751	.741	.870	.391	.359	.094	.188	.766	.734			15,730	14 700
09-	9/16 -12 UNC-3A	.562	.558	.876	.865	1.010	.438	.406	.109	.203	.891	.859	.078	.046	20 300	18 700
10-	5/8-11 UNC-3A	.624	.620	.939	.928	1,090	.484	.453	.140	.234	.954	.922			25 100	23 000
12-	3/4-10 UNC-3A	.749	.744	1.064	1.053	1.230	.578	.547	.187	.281	1.079	1.047			37 800	33 150

(a) GRIP LENGTH OF BOLTS SHALL BE MEASURED FROM THE UNDERSIDE OF THE HEAD TO END OF THE FULL CYLINDRICAL PORTION OF THE SHANK. COMPLETE THREADS SHALL BEGIN WITHIN TWO-THREAD PITCH MAXIMUM. TWO-THREAD PITCH MAXIMUM MAY CONSIST OF INCOMPLETE THREAD OR EXTRUSION ANGLE.

(b) REFERENCE DIMENSIONS ARE FOR DESIGN PURPOSES ONLY AND ARE NOT AN INSPECTION REQUIREMENT.
(c) MINIMUM YIELD STRENGTH = 76.7 PERCENT OF RATED TENSION.

MATERIAL: STEEL, SEE PROCUREMENT SPECIFICATIONS.

B PLATING: CADMIUM PLATE QQ-P-416, TYPE II, CLASS 2.

DIMENSIONS IN INCHES. UNLESS OTHERWISE SPECIFIED, TOLERANCES: DECIMALS ±.010 , ANGLES ± 5°.
EXAMPLES OF PART NUMBERS: MS20074-05-07 = 5/16-18 UNC-3A BOLT, .375 GRIP, .922 LONG.
UNASSIGNED DASH NUMBERS SHALL NOT BE USED.

THIS STANDARD TAKES PRECEDENCE OVER DOCUMENTS REFERENCED HEREIN.

REFERENCED DOCUMENTS SHALL BE OF THE ISSUE IN EFFECT ON DATE OF INVITATIONS FOR BID.

THIS DOCUMENT HAS BEEN PROMULGATED BY THE DEPARTMENT OF DEFENSE AS THE MILITARY STANDARD TO LIMIT THE SELECTION OF THE ITEM, PRODUCT, OR DESIGN COVERED HEREIN IN ENGINEERING, DESIGN, AND PROCUREMENT, THIS STANDARD SHALL BECOME EFFECTIVE NOT LATER THAN 90 DAYS AFTER THE LATEST DATE OF APPROVAL SHOWN.

APPROVED 16 SEPT 58 REVISED (A) 28 MAY 64 (B) 13 JUNE 1965

P. A.	INTERNATIONAL INTEREST	TITLE	MILITARY STANDARD
AIR FORCE - 82 Other Cust ARMY - AV NAVY - AS AIR FORCE - 99		BOLT, MACHINE, AIRCRAFT, DRILLED HEAD, COARSE THREAD	MS 20074
PROCUREMENT SPECIFICATION MIL-B-6812	SUPERSEDES	AN73 THRU AN81	SHEET 1 OF 3

MS20074 (AN-73 through AN-81) engine bolt specifications.

FED. SUP CLASS
5306

LENGTH DASH NO.	-03 GRIP +.016	-03 LENGTH +.031 -.016	-03 K REF (MIN)	-04 GRIP +.016	-04 LENGTH +.031 -.016	-04 K REF (MIN)	-05 GRIP ±.016	-05 LENGTH +.031 -.016	-05 K REF (MIN)	-06 GRIP +.016	-06 LENGTH +.031 -.016	-06 K REF (MIN)	-07 GRIP +.016	-07 LENGTH +.031 -.016	-07 K REF (MIN)
02	.062	.344	.250												
03	.062	..469	.375	.062	.469	.375									
04	.062	.594	.500	.062	.594	.500	.062	.609	.515						
05	.125	.656	.499	.125	.656	.499	.125	.672	.515	.062	.734	.640			
06	.250	.781	.499	.250	.781	.499	.250	.797	.515	.125	.797	.640	.062	.797	.703
07	.375	.906	.499	.375	.906	.499	.375	.922	.515	.250	.922	.640	.188	.922	.702
10	.500	1.031	.499	.500	1.031	.499	.500	1.047	.515	.375	1.047	.640	.312	1.047	.703
11	.625	1.156	.499	.625	1.156	.499	.625	1.172	.515	.500	1.172	.640	.438	1.172	.702
12	.750	1.281	.499	.750	1.281	.499	.750	1.297	.515	.625	1.297	.640	.562	1.297	.703
13	.875	1.406	.499	.875	1.406	.499	.875	1.422	.515	.750	1.422	.640	.688	1.422	.702
14	1.000	1.531	.499	1.000	1.531	.499	1.000	1.547	.515	.875	1.547	.640	.812	1.547	.703
15	1.125	1.656	.499	1.125	1.656	.499	1.125	1.672	.515	1.000	1.672	.640	.938	1.672	.702
16	1.250	1.781	.499	1.250	1.781	.499	1.250	1.797	.515	1.125	1.797	.640	1.062	1.797	.703
17	1.375	1.906	.499	1.375	1.906	.499	1.375	1.922	.515	1.250	1.922	.640	1.188	1.922	.702
20	1.500	2.031	.499	1.500	2.031	.499	1.500	2.047	.515	1.375	2.047	.640	1.312	2.047	.703
21	1.625	2.156	.499	1.625	2.156	.499	1.625	2.172	.515	1.500	2.172	.640	1.438	2.172	.702
22	1.750	2.281	.499	1.750	2.281	.499	1.750	2.297	.515	1.625	2.297	.640	1.562	2.297	.703
23	1.875	2.406	.499	1.875	2.406	.499	1.875	2.422	.515	1.750	2.422	.640	1.688	2.422	.702
24	2.000	2.531	.499	2.000	2.531	.499	2.000	2.547	.515	1.875	2.547	.640	1.812	2.547	.703
25	2.125	2.656	.499	2.125	2.656	.499	2.125	2.672	.515	2.000	2.672	.640	1.938	2.672	.702
26	2.250	2.781	.499	2.250	2.781	.499	2.250	2.797	.515	2.125	2.797	.640	2.062	2.797	.703
27	2.375	2.906	.499	2.375	2.906	.499	2.375	2.922	.515	2.250	2.922	.640	2.188	2.922	.702
30	2.500	3.031	.499	2.500	3.031	.499	2.500	3.049	.515	2.375	3.049	.640	2.312	3.049	.703
31	2.625	3.156	.499	2.625	3.156	.499	2.625	3.172	.515	2.500	3.172	.640	2.438	3.172	.702
32	2.750	3.281	.499	2.750	3.281	.499	2.750	3.297	.515	2.625	3.297	.640	2.562	3.297	.703
33	2.875	3.406	.499	2.875	3.406	.499	2.875	3.422	.515	2.750	3.422	.640	2.688	3.422	.702
34	3.000	3.531	.499	3.000	3.531	.499	3.000	3.247	.515	2.875	3.547	.640	2.812	3.547	.703
35	3.125	3.656	.499	3.125	3.656	.499	3.125	3.672	.515	3.000	3.672	.640	2.938	3.672	.702
36	3.250	3.781	.499	3.250	3.781	.499	3.250	3.797	.515	3.125	3.797	.640	3.062	3.797	.703
37	3.375	3.906	.499	3.375	3.906	.499	3.375	3.922	.515	3.250	3.922	.640	3.188	3.922	.702
40	3.500	4.031	.499	3.500	4.031	.499	3.500	4.047	.515	3.375	4.047	.640	3.312	4.047	.703
41	3.625	4.156	.499	3.625	4.156	.499	3.625	4.172	.515	3.500	4.172	.640	3.438	4.172	.702
42	3.750	4.281	.499	3.750	4.281	.499	3.750	4.297	.515	3.625	4.297	.640	3.562	4.297	.703
43	3.875	4.406	.499	3.875	4.406	.499	3.875	4.422	.515	3.750	4.422	.640	3.688	4.422	.702
44	4.000	4.531	.499	4.000	4.531	.499	4.000	4.547	.515	3.875	4.547	.640	3.812	4.547	.703
45	4.125	4.656	.499	4.125	4.656	.499	4.125	4.672	.515	4.000	4.672	.640	3.938	4.672	.702
46	4.250	4.781	.499	4.250	4.781	.499	4.250	4.797	.515	4.125	4.797	.640	4.062	4.797	.703
47	4.375	4.906	.499	4.375	4.906	.499	4.375	4.922	.515	4.250	4.922	.640	4.188	4.922	.702
50	4.500	5.031	.499	4.500	5.031	.499	4.500	5.047	.515	4.375	5.047	.640	4.312	5.047	.703
51	4.625	5.156	.499	4.625	5.156	.499	4.675	5.172	.515	4.500	5.172	.640	4.438	5.172	.702
52	4.750	5.281	.499	4.750	5.281	.499	4.750	5.297	.515	4.625	5.297	.640	4.562	5.297	.703
53	4.875	5.406	.499	4.875	5.406	.499	4.875	5.422	.515	4.750	5.422	.640	4.688	5.422	.702
54	5.000	5.531	.499	5.000	5.531	.499	5.000	5.547	.515	4.875	5.547	.640	4.812	5.547	.703
55	5.125	5.656	.499	5.125	5.656	.499	5.125	5.672	.515	5.000	5.672	.640	4.938	5.672	.702
56	5.250	5.781	.499	5.250	5.781	.499	5.250	5.797	.515	5.125	5.797	.640	5.062	5.797	.703
57	5.375	5.906	.499	5.375	5.906	.499	5.375	5.922	.515	5.250	5.922	.640	5.188	5.922	.702
60	5.500	6.031	.499	5.500	6.031	.499	5.500	6.047	.515	5.375	6.047	.640	5.312	6.047	.703

APPROVED 18 SEP 64 REVISED (B) FOR CHANGES SEE SHEETS 1 AND 2

P.A. AIR FORCE 82 Other Cust ARMY AV NAVY AS AIR FORCE 99	INTERNATIONAL INTEREST	TITLE BOLT, MACHINE, AIRCRAFT, DRILLED HEAD, COARSE THREAD	MILITARY STANDARD MS 20074
PROCUREMENT SPECIFICATION MIL-B-6812	SUPERSEDES:		SHEET 2 OF 8

MS20074 engine bolt specifications.

LENGTH DASH NO.	-08			-09			-10			-12		
	GRIP ±.016	LENGTH +.031 -.016	K REF (MIN)	GRIP ±.016	LENGTH +.031 -.016	K REF (MIN)	GRIP ±.016	LENGTH +.031 -.016	K REF (MIN)	GRIP ±.016	LENGTH +.031 -.016	K REF (MIN)
10	.250	1.047	.765									
11	.375	1.172	.765									
12	.500	1.297	.765	.438	1.312	.842						
13	.625	1.422	.765	.562	1.438	.844						
14	.750	1.547	.765	.688	1.562	.842	.562	1.562	.968			
15	.875	1.672	.765	.812	1.688	.844	.688	1.688	.968			
16	1.000	1.797	.765	.938	1.812	.842	.812	1.812	.968	.750	1.812	1.030
17	1.125	1.922	.765	1.062	1.938	.844	.938	1.938	.968	.875	1.938	1.031
20	1.250	2.047	.765	1.188	2.062	.842	1.062	2.062	.968	1.000	2.062	1.030
21	1.375	2.172	.765	1.312	2.188	.844	1.188	2.188	.968	1.125	2.188	1.031
22	1.500	2.297	.765	1.438	2.312	.842	1.312	2.312	.968	1.250	2.312	1.030
23	1.625	2.422	.765	1.562	2.438	.844	1.438	2.438	.968	1.375	2.438	1.031
24	1.750	2.547	.765	1.688	2.562	.842	1.562	2.562	.968	1.500	2.562	1.030
25	1.875	2.672	.765	1.812	2.688	.844	1.688	2.688	.968	1.675	2.688	1.031
26	2.000	2.797	.765	1.938	2.812	.842	1.812	2.812	.968	1.750	2.812	1.030
27	2.125	2.922	.765	2.062	2.938	.844	1.938	2.938	.968	1.875	2.938	1.031
30	2.250	3.047	.765	2.188	3.062	.842	2.062	3.062	.968	2.000	3.062	1.030
31	2.375	3.172	.765	2.312	3.188	.844	2.188	3.188	.968	2.125	3.188	1.031
32	2.500	3.297	.765	2.438	3.312	.842	2.312	3.312	.968	2.250	3.312	1.030
33	2.625	3.422	.765	2.562	3.438	.844	2.438	3.438	.968	2.375	3.438	1.031
34	2.750	3.547	.765	2.688	3.562	.842	2.562	3.562	.968	2.500	3.562	1.030
35	2.875	3.672	.765	2.812	3.688	.844	2.688	3.688	.968	2.625	3.688	1.031
36	3.000	3.797	.765	2.938	3.812	.842	2.812	3.812	.968	2.750	3.812	1.030
37	3.125	3.922	.765	3.062	3.938	.844	2.938	3.938	.968	2.875	3.938	1.031
40	3.250	4.047	.765	3.188	4.062	.842	3.062	4.062	.968	3.000	4.062	1.030
41	3.375	4.172	.765	3.312	4.188	.844	3.188	4.188	.968	3.125	4.188	1.031
42	3.500	4.297	.765	3.438	4.312	.842	3.312	4.312	.968	3.250	4.312	1.030
43	3.675	4.422	.765	3.562	4.438	.844	3.438	4.438	.968	3.375	4.438	1.031
44	3.750	4.547	.765	3.688	4.562	.842	3.562	4.562	.968	3.500	4.562	1.030
45	3.875	4.672	.765	3.812	4.688	.844	3.688	4.688	.968	3.625	4.688	1.031
46	4.000	4.797	.765	3.938	4.812	.842	3.812	4.812	.968	3.750	4.812	1.030
47	4.125	4.922	.765	4.062	4.938	.844	3.938	4.938	.968	3.875	4.938	1.031
50	4.250	5.047	.765	4.188	5.062	.842	4.062	5.062	.968	4.000	5.062	1.030
51	4.375	5.172	.765	4.312	5.188	.844	4.188	5.188	.968	4.125	5.188	1.031
52	4.500	5.297	.765	4.438	5.312	.842	4.312	5.312	.968	4.250	5.312	1.030
53	4.625	5.422	.765	4.562	5.438	.844	4.438	5.438	.968	4.375	5.438	1.031
54	4.750	5.547	.765	4.688	5.562	.842	4.562	5.562	.968	4.500	5.562	1.030
55	4.875	5.672	.765	4.812	5.688	.844	4.688	5.688	.968	4.625	5.688	1.031
56	5.000	5.797	.765	4.938	5.812	.842	4.812	5.812	.968	4.750	5.812	1.030
57	5.125	5.922	.765	5.062	5.938	.844	4.938	5.938	.968	4.875	5.938	1.031
60	5.250	6.047	.765	5.188	6.016	.842	5.062	6.062	.968	5.000	6.062	1.030

MS20074 engine bolt specifications.

If you want to go wild on the weight saving bit without going to titanium (which galls on steel anyway), you can use the NAS-1103 through NAS-1120 series of shear bolts. These are rated at 95,000 psi ultimate shear strength. The material is rated at 160,000 psi ultimate tensile strength but that is only of academic interest, as they are designed for shear applications exclusively. They have thinner heads and shorter thread lengths than the tension bolts and are used with shear nuts, often called thin or ½ high nuts. A close equivalent is the NAS-464 series.

The current trend in race car construction is toward the use of internal wrenching bolts. There are a couple of reasons for this. The first is that most of our road racing cars come from England, and commercial socket-head cap screws are the only good bolts that are readily obtainable there. The second reason is that space for bolt heads and the wrenches to tighten them is always at a premium, and the internal wrenching bolt requires less of it. The NAS-144 through NAS-159 series or the MS20004 through MS20024 series come in standard AN grip lengths and UNRF threads. Rated at 160,000 psi ultimate tensile strength and 96,000 psi ultimate shear strength, they are vastly superior to the commercial items in terms of fatigue resistance.

The NAS bolts feature generous fillet radii between the bolt head and the shank and so require the use of NAS-1430 or MS20002 series chamfered washers. The strength of the aerospace bolts is obtained from alloying and heat treatment, so the bolts retain high levels of toughness and resistance to fatigue. The strength of commercial socket-head cap screws is a function of hardness—at the cost of decreased toughness and lessened resistance to fatigue. Again, the FAA won't allow a commercial bolt on an aircraft.

If you are looking for really high strength, or if the designer has erred and you are bending an AN bolt in service, use the NAS-624 through NAS-644 series of 180,000 psi twelve-point external wrenching bolts. Although higher strength bolts are available, I feel that these bolts are as good as we have a need for in a chassis. They are not hard to find and are sometimes available on the surplus market. They also require a beveled washer.

All SAE grades and all aerospace bolts are stamped with identifying marks on the bolt heads. Unfortunately, the commercial socket-headed cap screws are often not marked, so to be safe, you have to know the manufacturer. This means that you cannot buy surplus and that you have to both trust the supplier and see the box, when you are buying retail.

Bolt Talk Three: Finding the right stuff

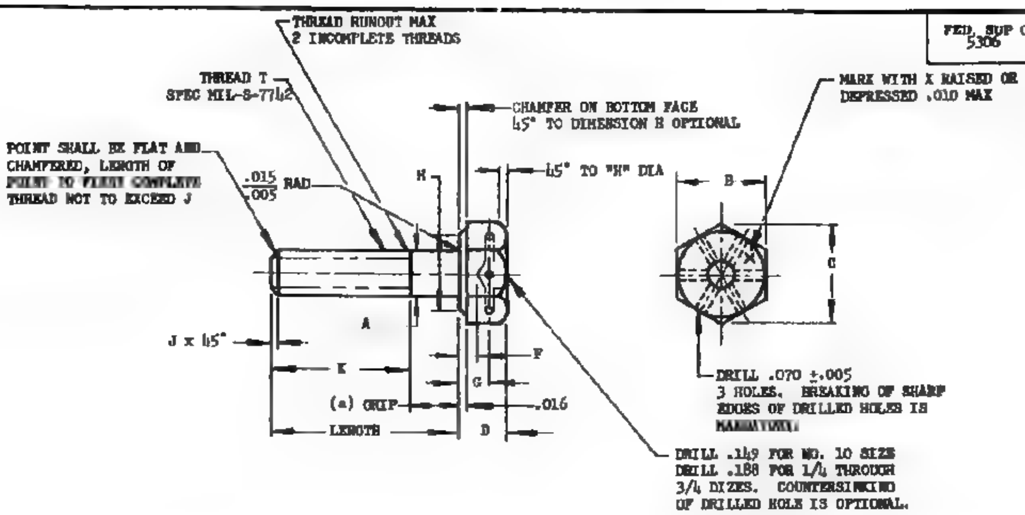
Finding a reasonable source for top-quality threaded fasteners requires some detective work. Nationwide firms that stock aerospace hardware are Van Deusen Aircraft Supply and Albany Products. But the price is high and they don't like selling in small quantities. Earl's Performance Products and Aircraft Spruce are both very cooperative, however, and will sell in any quantity, and ship by UPS. For those of you in the Northeast, Charlie Vogelsang at the Dillsburg Aeroplane Works, Dillsburg, Pennsylvania (see appendices for address), stocks a complete line of AN hardware, tubing and sheet metal. He ships same day UPS and can save your life. God bless and prosper him. Your local general aviation airport probably has a supply store that

Shear and tensile strength of AN-73 through AN-80 (UNC) bolts (Same as MS20073 through MS20080)
Source: ANC-5

AN #	Diameter (in.)	Allowable single shear load (lb.) @ full diam.	Allowable tensile load (lb.) @ thread root diam.	Area of cross section (full diam.)	Stress area UNF (sq. in.)
AN-73	3/16	2,070	1,800	0.0276	0.0174
AN-74	1/4	3,680	3,360	0.0491	0.0317
AN-75	5/16	5,750	5,660	0.0767	0.0522
AN-76	3/8	8,280	8,470	0.1105	0.0773
AN-77	7/16	11,200	11,680	0.1503	0.1060
AN-78	1/2	12,760	15,730	0.1963	0.1416
AN-79	9/16	18,700	20,300	0.2485	0.1816
AN-80	5/8	23,000	25,100	0.3068	0.2256

Shear and tensile strength of AN-73 through AN-80 (UNC) engine bolts (Identical to MS20073 through MS20080).

FED. SUP CLASS. 5306



SIZE DASH NO.	THREAD T	A DIA		B		(b) C REF	D		F MIN	G +.000 -.010	H DIA		J		RATED STRENGTH, LBS (c) TENSION AT ROOT DIA	SINGLE SHEAR AT FULL DIA
		MAX	MIN	MAX	MIN		MAX	MIN			MAX	MIN	MAX	MIN		
03-	NO. 10-32 NF-3A	.189	.186	.376	.367	[illegible]	.203	.172	.039	.120	.391	.359	.047	.015	2 210	2 125
04-	1/4-28 UNF-3A	.249	.246	.439	.430	[illegible]	.234	.203	.052	.144	.454	.422	.047	.015	4 080	3 680
05-	5/16-24 UNF-3A	.312	.309	.501	.492	.580	.297	.266	.078	.172	.516	.484	.063	.031	6 500	5 750
06-	3/8-24 UNF-3A	.374	.371	.564	.554	[illegible]	.297	.266	.078	.172	.579	.547	.063	.031	10 100	8 800
07-	7/16-20 UNF-3A	.437	.433	.689	.679	.790	.344	.313	.078	.172	.704	.672	.063	.031	13 600	11 250
08-	1/2-20 UNF-3A	.499	.495	.751	.741	.870	.391	.359	.094	.188	.766	.734	.063	.031	18 500	14 700
09-	9/16-18 UNF-3A	.562	.558	.876	.865	1.010	.438	.406	.109	.203	.891	.859	.078	.046	23 600	18 700
10-	5/8-18 UNF-3A	.624	.620	.939	.928	1.090	.484	.453	.140	.234	.954	.922	.078	.046	30 100	23 000
12-	3/4-16 UNF-3A	.749	.744	1.064	1.053	1.230	.578	.547	.187	.281	1.079	1.047	.078	.046	44 000	33 150

(a) GRIP LENGTH OF BOLTS SHALL BE MEASURED FROM THE UNDERSIDE OF THE HEAD TO THE END OF THE FULL CYLINDRICAL PORTION OF THE SHANK. COMPLETE THREADS SHALL BEGIN WITHIN TWO-THREAD PITCH MAXIMUM. TWO-THREAD PITCH MAXIMUM MAY CONSIST OF INCOMPLETE THREAD OR EXTRUSION ANGLE.

(b) REFERENCE DIMENSIONS ARE FOR DESIGN PURPOSES ONLY AND ARE NOT AN INSPECTION REQUIREMENT.

(c) MINIMUM YIELD STRENGTH = 76.7 PERCENT OF RATED TENSION.

MATERIAL: STEEL, SEE PROCUREMENT SPECIFICATION.

PLATING: CADMIUM PLATE QQ-P-416, TYPE 2, CLASS 3.

DIMENSIONS IN INCHES. UNLESS OTHERWISE SPECIFIED, TOLERANCES: DECIMALS ±.010, ANGLES ±5°.

EXAMPLE OF PART NUMBERS: MS20073-05-07 = 5/16-24 UNF-3A BOLT, .375 GRIP, .922 LONG.

[redacted]

THE ACROSS FLAT DIMENSIONS OF THE HEXAGON HEADS ARE IN AGREEMENT WITH AMERICAN-BRITISH-CANADIAN AIR STD 17/2.

INTERCHANGEABILITY RELATION WITH AN73 THROUGH AN81 PARTS: MS20073 (ASG) PARTS AND AN73 THROUGH AN81 PARTS OF LIKE THREAD AND GRIP LENGTHS ARE UNIVERSALLY FUNCTIONALLY AND DIMENSIONALLY INTERCHANGEABLE.

THIS STANDARD TAKES PRECEDENCE OVER DOCUMENTS REFERENCED HEREIN.

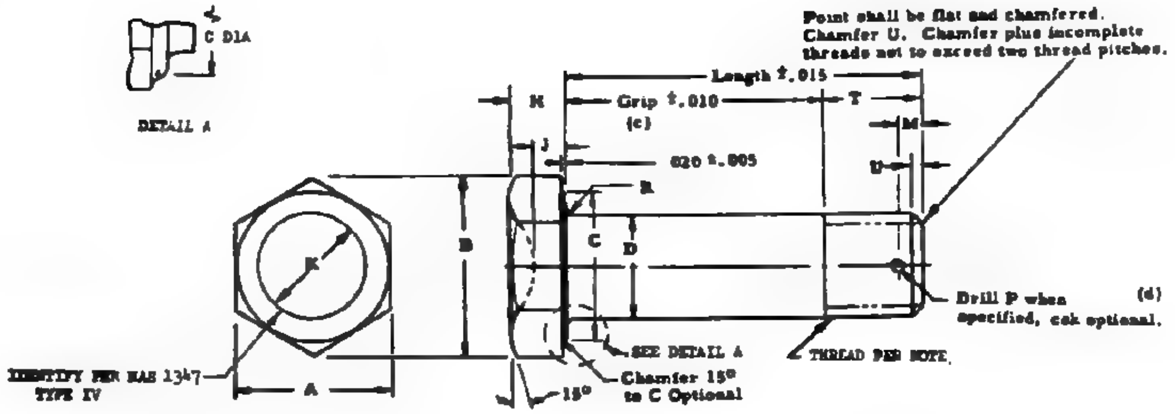
REFERENCED DOCUMENTS SHALL BE OF THE ISSUE IN EFFECT ON DATE OF INVITATIONS FOR BID.

APPROVED 16 Sep 58 REVISED

THIS DOCUMENT HAS BEEN PROMULGATED BY THE DEPARTMENT OF DEFENSE AS THE MILITARY STANDARD TO LIMIT THE SELECTION OF THE ITEM, PRODUCT, OR DESIGN COVERED HEREIN IN ENGINEERING, DESIGN, AND PROCUREMENT. THIS STANDARD SHALL BECOME EFFECTIVE NOT LATER THAN 90 DAYS AFTER THE LATEST DATE OF APPROVAL SHOWN.

CUSTODIANS	OTHER INT.	MILITARY STANDARD	MS20073 (ASG)
Navy - BuAer Air Force	A - N - AF -	BOLT, MACHINE, AIRCRAFT, DRILLED HEAD, FINE THREAD	
PROCUREMENT SPECIFICATION MIL-B-6812		SUPERSEDES: AN73 THRU AN81	SHEET 1 OF 5

MS20073 (AN-73 through AN-81) engine bolt specifications.



Basic Series	THREAD UNJF-3A SEE THD NOTE	A	B Ref	C Min (g)	D DIA Before Plate	D DIA After Plate	H +.015 -.000	J +.015 -.000	K Dia ±.01	M ±.010	P (4) Dia +.010 -.000	R Rad
NAS1103	.1900-32	.376 .367	.43	.335	.1887 .1881	.1895 .1865	.110	.073	.19	.117	.070	.020 .010
NAS1104	.2500-28	.439 .430	.51	.398	.2487 .2481	.2495 .2485	.125	.083	.25	.116	.076	.020 .010
NAS1105	.3125-24	.502 .492	.58	.460	.3112 .3106	.3120 .3110	.156	.104	.31	.119	.076	.020 .010
NAS1106	.3750-24	.564 .553	.65	.523	.3737 .3731	.3745 .3735	.188	.125	.38	.120	.106	.025 .015
NAS1107	.4375-20	.690 .679	.79	.648	.4362 .4356	.4370 .4360	.219	.146	.44	.124	.106	.025 .015
NAS1108	.5000-20	.752 .741	.87	.710	.4987 .4981	.4995 .4985	.250	.167	.50	.123	.106	.030 .020
NAS1109	.5625-18	.877 .865	1.01	.835	.5607 .5601	.5615 .5605	.281	.188	.56	.124	.141	.035 .020
NAS1110	.6250-18	.940 .928	1.09	.898	.6232 .6226	.6240 .6230	.312	.208	.62	.124	.141	.040 .025
NAS1112	.7500-16	1.064 1.052	1.23	1.023	.7482 .7476	.7490 .7480	.375	.250	.75	.128	.141	.045 .030
NAS1114	.8750-14	1.252 1.239	1.44	1.210	.8732 .8726	.8740 .8730	.438	.292	.88	.134	.141	.050 .035
NAS1116	1.0000-12	1.440 1.427	1.66	1.398	.9982 .9976	.9990 .9980	.500	.333	1.00	.139	.141	.060 .045
NAS1118	1.1250-12	1.627 1.614	1.88	1.585	1.1232 1.1221	1.1240 1.1225	.562	.375	1.12	.170	.141	.070 .055
NAS1120	1.2500-12	1.815 1.801	2.10	1.772	1.2482 1.2471	1.2490 1.2475	.625	.417	1.25	.170	.141	.075 .060

THREADS THREADS PER MIL-S-8879 EXCEPT MAJOR DIA TO BE A MIN OF .001 BELOW MIN SHANK DIA. THREAD RUNOUT PER MIL-B-7838. ~~BOLT MANUFACTURER~~ [illegible] ~~BOLTS WITH MIL-S-7742 THREADS MAY BE USED UNTIL STOCK IS DEPLETED.~~

LIST OF CURRENT SHEETS

(8)

NO.	REV.
1	8
2	5
3	4

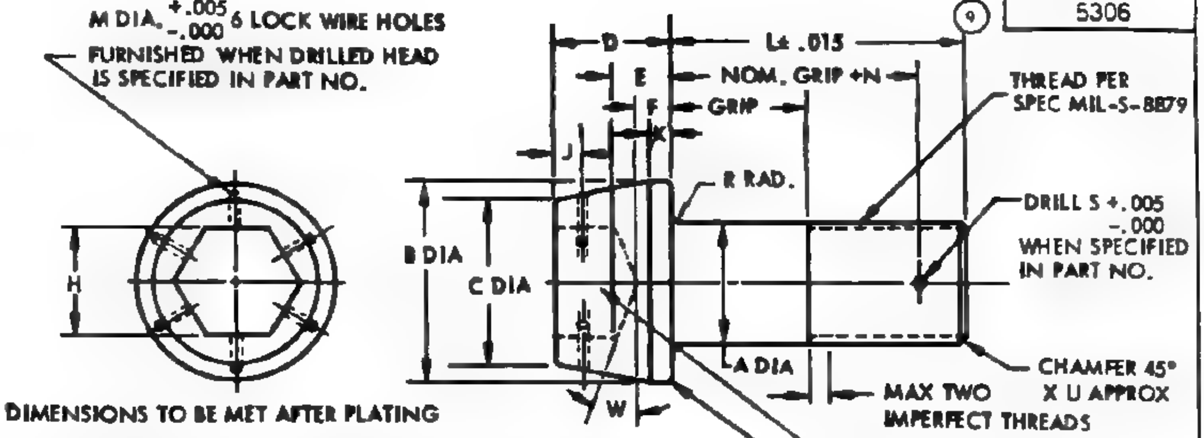
(8) INACTIVE FOR DESIGN AFTER JULY 1, 1976
SEE NAS6203 THRU NAS6220

CUSTODIAN. NATIONAL AEROSPACE STANDARDS COMMITTEE

PROCUREMENT SPECIFICATION	TITLE	CLASSIFICATION
SEE ABOVE	BOLT, SHEAR – HEXAGON HEAD MODIFIED, SHORT THREAD	STANDARD PART NAS 1103 THRU 1120 SHEET 1 OF 3

NAS–464 shear bolt chart.

FED SUP CLASS
5306



LOCK WIRE HOLES MUST BE FREE FROM BURRS AND SHARP EDGES.
LETTER "R" ON BOLT DESIGNATES THAT THREADS HAVE BEEN ROLLED AFTER HEAT TREATMENT SEE PARAGRAPH 4.1.1 OF NAS 159.

BASIC NAS PART NO.	THREAD	A	B	C	D	E	F MIN.	H	J	K	M	N	R	S	U	W MAX.
NAS 144	1/4-28 UNJF-3A	.2490 / .2460	.438 / .430	.381	.226	.122	.072	.2210 / .2190	.063	.063	.046	11/32	.041 / .026	.076	.03	20°
NAS 145	5/16-24 UNJF-3A	.3115 / .3085	.531 / .523	.501	.276	.142	.075	.3150 / .3130	.063	.063	.070	25/64	.041 / .026	.076	.05	20°
NAS 146	3/8-24 UNJF-3A	.3740 / .3710	.625 / .615	.583	.343	.170	.091	.3785 / .3755	.094	.063	.070	15/32	.057 / .042	.106	.05	20°
NAS 147	7/16-20 UNJF-3A	.4365 / .4330	.750 / .740	.594	.407	.209	.130	.3785 / .3755	.094	.063	.070	33/64	.057 / .042	.106	.05	20°
NAS 148	1/2-20 UNJF-3A	.4990 / .4955	.813 / .802	.751	.460	.223	.130	.5035 / .5005	.094	.063	.070	5/8	.057 / .042	.106	.05	18°
NAS 149	9/16-18 UNJF-3A	.5615 / .5575	.936 / .927	.775	.529	.261	.160	.5035 / .5005	.125	.094	.070	43/64	.057 / .042	.141	.06	20°
NAS 150	5/8-18 UNJF-3A	.6240 / .6200	1.000 / .988	.850	.595	.288	.175	.5660 / .5630	.188	.094	.070	3/4	.073 / .058	.141	.06	20°
NAS 152	3/4-16 UNJF-3A	.7490 / .7445	1.188 / 1.174	.938	.732	.357	.232	.6297 / .6257	.188	.094	.070	7/8	.073 / .058	.141	.06	20°
NAS 154	7/8-14 UNJF-3A	.8740 / .8690	1.438 / 1.424	1.098	.846	.406	.259	.7547 / .7507	.188	.094	.070	31/32	.073 / .058	.141	.08	20°
NAS 156	1-14 UNJF-3A	.9990 / .9935	1.625 / 1.609	1.405	.952	.434	.259	1.0050 / 1.0010	.188	.094	.070	1-1/8	.073 / .058	.141	.08	18.5°
NAS 158	1-1/8-12 UNJF-3A	1.1240 / 1.1180	1.875 / 1.857	1.423	1.091	.512	.320	1.0050 / 1.0010	.188	.094	.070	1-3/16	.073 / .058	.141	.09	20°

for 1-1/4, 1-3/8 & 1-1/2 DIA. BOLTS, SEE NAS172, NAS174 & NAS176.

CODE: ADD "DH" TO PART NUMBER TO DESIGNATE DRILLED HEAD.
ADD "A" TO PART NUMBER TO DESIGNATE DRILLED SHANK.
FOR SUFFIX DASH NUMBER TO DESIGNATE GRIP AND LENGTH, SEE PAGE 2

EXAMPLES: NAS144-25 = 1/4-28 BOLT, INT. WR. 1-9/16 LONG, UNDRILLED.
NAS144DH-25 = 1/4-28 BOLT, INT. WR. 1-9/16 LONG, DRILLED HEAD.
NAS144ADH-25 = 1/4-28 BOLT, INT. WR. 1-9/16 LONG, DRILLED HEAD, DRILLED SHANK.

MATERIAL: ALLOY STEEL IN ACCORDANCE WITH PROCUREMENT SPECIFICATION.

HEAT TREAT: 160,000 TO 180,000 PSI TENSIL STRENGTH. SEE PROCUREMENT SPECIFICATION FOR DETAILS.

FINISH: CADMIUM PLATING PER QQ-P-416 TYPE II, CLASS 2. PARTS WITH CLASS 3 PLATING MAY BE FURNISHED FROM SUPPLIER'S STOCK UNTIL 15 SEPT. 1975. BAKE AFTER PLATING IN ACCORDANCE WITH PROCUREMENT SPECIFICATION.

LIMITS: UNLESS OTHERWISE SPECIFIED, ±.010.

DIMENSIONS IN INCHES.

ENGR. REF.: APPLICABLE WASHERS, MS20002.

ENTIRE SHEET REVISED, NAS144 THRU NAS158 FORMERLY PUBLISHED AS 11 SEPARATE SHEETS.

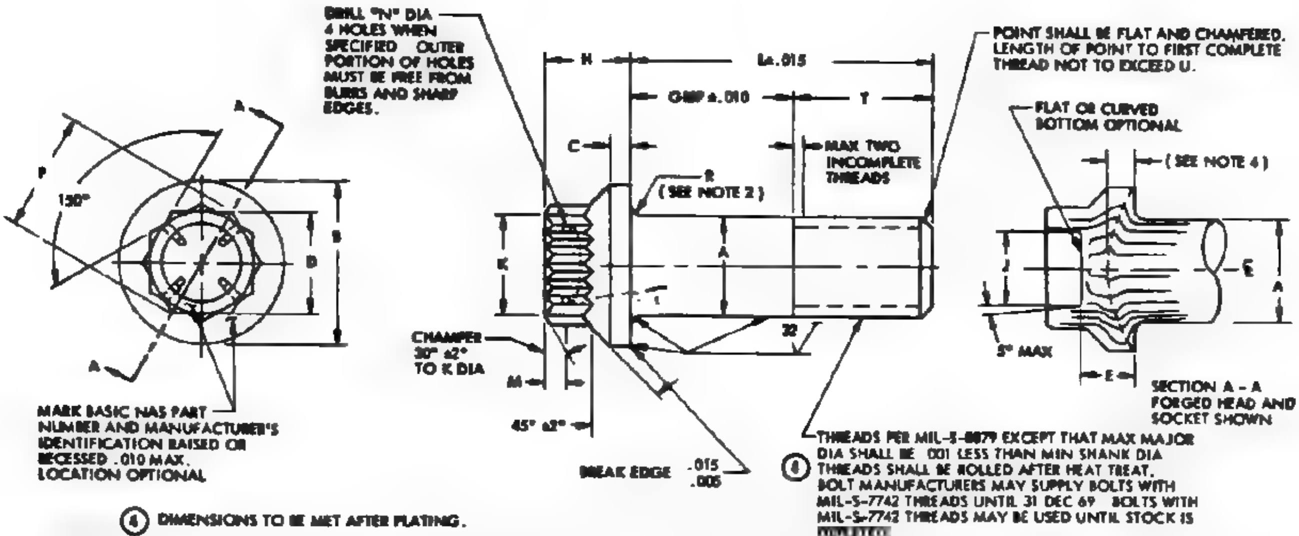
LIST OF CURRENT SHEETS

NO.	REV.
1	9
2	6

9 INACTIVE FOR NEW DESIGN AFTER OCTOBER 1. 1986 USE MS 20004 thru MS - 20024

CUSTODIAN	NATIONAL AEROSPACE STANDARDS COMMITTEE	
PROCUREMENT SPECIFICATION NAS159	TITLE BOLT–INTERNAL WRENCHING–STEEL 1/4–28 THRU 1–1/8–12	CLASSIFICATION STANDARD PART NAS144 THRU 158 SHEET 1 OF 2

NAS–144 through NAS–159 internal wrenching bolt chart.



(4) DIMENSIONS TO BE MET AFTER PLATING.

BASIC PART NUMBER	THREAD UNJF-3A	A (a)	B (a) ±.005	C ±.010	D (a)	E MIN	H ±.010	J +.010 -.030	K ±.010	M ±.011	N ±.005	P (b) REF	R	T (b) REF	U	[redacted] (a) X	Y	Z	ULTIMATE TENSILE LBS, MIN	WEIGHT - LBS
NAS624	.2500-28	.2495 .2485	.433	.069	.315 .306	.135	.300	.180	.312	.062	.037	.360	.041 .031	.485	.05	.005	.004	.0045	6,960	.0128+.0139 X GRIP LG
NAS625	.3125-24	.3120 .3110	.526	.082	.377 .368	.162	.348	.215	.375	.070	.055	.431	.041 .031	.577	.05	.006	.005	.0045	11,050	.0221+.0217 X GRIP LG
NAS626	.3750-24	.3745 .3735	.644	.091	.439 .430	.197	.388	.260	.438	.070	.055	.502	.057 .047	.619	.06	.008	.006	.0045	17,100	.0392+.0313 X GRIP LG
NAS627	.4375-20	.4370 .4360	.745	.099	.503 .494	.228	.435	.320	.[redacted]	.070	.055	.576	.057 .047	.690	.06	.009	.008	.0060	23,375	.0603+.0426 X GRIP LG
NAS6[redacted]	.5000-20	.4995 .4985	.873	.123	.565 .556	.254	.504	.380	.567	.094	.055	.648	.057 .047	.741	.08	.010	.008	.0060	30,825	.0824+.0556 X GRIP LG
NAS[redacted]	.5625-18	.5615 .5605	.933	.133	.628 .619	.287	.557	.440	.625	.094	.055	.720	.057 .047	.826	.08	.011	.009	.0060	39,250	.1148+.0708 X GRIP LG
NAS[redacted]	.6250-18	.6240 .6230	1.045	.150	.691 .682	.327	.618	.500	.688	.094	.055	.704	.073 .063	.871	.09	.012	.010	.0060	49,050	.1475+.0871 X GRIP LG
NAS[redacted]	.7500-16	.7490 .7480	1.225	.178	.817 .808	.380	.711	.570	.813	.094	.055	.939	.073 .063	.993	.09	.015	.012	.0060	71,100	.2308+.1255 X GRIP LG
NAS634	.8750-14	.8740 .8730	1.433	.198	.943 .934	.438	.808	.650	.938	.125	.055	1.081	.073 .063	1.194	.10	.018	.014	.0090	96,860	.3462+.1708 X GRIP LG
NAS636 / NAS636A*	1.0000-12 / 1.0000-14	.9990 .9980	1.620	.222	1.066 1.056	.493	.923	.740	1.062	.125	.055	1.228	.073 .063	1.382	.10	.020	.016	.0090	126,250 / 128,550	.5200+.2231 X GRIP LG
NAS638	1.1250-12	1.1240 1.1225	1.870	.258	1.255 1.245	.556	1.051	.840	1.250	.125	.055	1.434	.073 .063	1.587	.13	.022	.013	.0090	162,000	.8703+.2798 X GRIP LG
NAS640	1.2500-12	1.2490 1.2475	2.120	.269	1.318 1.308	.636	1.155	.950	1.312	.125	.055	1.519	.089 .077	1.697	.13	.025	.021	.0090	202,500	1.421+.3485 X GRIP LG
NAS642	1.3750-12	1.3740 1.3725	2.308	.274	1.444 1.434	.698	1.266	1.070	1.438	.188	.055	1.660	.089 .077	1.822	.13	.028	.023	.012[redacted]	246,375	1.950 +.4249 X GRIP LG
NAS644	1.5000-12	1.4990 1.4975	2.495	.335	1.632 1.622	.750	1.434	1.200	1.625	.188	.055	1.879	.089 .077	1.947	.13	.030	.023	.0120	295,875	2.459 +.5418 X GRIP LG

* NS-3A

(a) DIMENSIONS A, B AND J SHALL BE CONCENTRIC TO EACH OTHER WITHIN VALUES SPECIFIED FOR X TOTAL INDICATOR READING: DIMENSIONS D AND J SHALL BE CONCENTRIC TO EACH OTHER WITHIN VALUES SPECIFIED FOR Y TOTAL INDICATOR READING. DIMENSION A AND THREAD PITCH DIAMETER SHALL BE CONCENTRIC TO EACH OTHER WITHIN VALUES SPECIFIED FOR Z TOTAL INDICATOR READING.

(b) REFERENCE DIMENSIONS ARE FOR DESIGN PURPOSES ONLY AND ARE NOT AN INSPECTION REQUIREMENT.

FATIGUE LOADING (POUNDS)		
BASIC PART NO.	LOW TENSION LOAD 10%	HIGH TENSION LOAD 100%
NAS624	282	2,820
NAS625	454	4,540
NAS626	731	7,310
NAS627	944	9,440
NAS628	1,290	12,900
NAS629	1,640	16,400
NAS630	2,050	20,500
NAS632	3,040	30,400
NAS634	4,160	41,600
NAS636	5,424	54,240
NAS636A	5,600	56,000
NAS638	7,030	70,300
NAS640	8,870	88,700
NAS642	10,900	109,000
NAS644	13,200	132,000

CODE: PREFIX DASH NUMBER WITH "H" TO DESIGNATE DRILLED HEAD BOLT.

EXAMPLE: NAS624-8 — .500 GRIP; (4) .2500-28 UNJF-3A THREAD

NAS628H13 — .812 GRIP; DRILLED HEAD; .5000-20 UNJF-3A THREAD

PROCUREMENT SPECIFICATION: NAS496 EXCEPT FOR HEAT TREAT, PLATING, MECHANICAL PROPERTIES AND FATIGUE REQUIREMENTS.

MATERIAL: ALLOY STEEL. SEE PROCUREMENT SPECIFICATION. THE USE OF 8735 STEEL IS RESTRICTED TO BOLTS WITH SHANK DIAMETERS SMALLER THAN 1 INCH.

HEAT TREAT: 180,000 TO 200,000 PSI TENSILE STRENGTH PER SPECIFICATION MIL-H-6875. ROCKWELL HARDNESS C39-C43.

FINISH: CADMIUM PLATE PER SPECIFICATION NAS672.

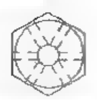
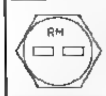
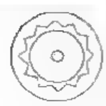
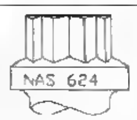
NOTES:
1. THESE BOLTS ARE SUITABLE FOR APPLICATIONS REQUIRING BOLTS WITH HIGH TENSILE-FATIGUE STRENGTH.
2. HEAD TO SHANK FILLET RADII SHALL BE COLD WORKED.
3. LIGHTENING HOLE IN HEAD MAY BE DRILLED OR FORMED.
4. MACROSTRUCTURE OF FORGED HEADS SHALL CONFORM TO THE GRAIN FLOW PATTERN SHOWN. THE INTERSECTION OF THE LONGITUDINAL AXIS OF THE BOLT AND THE APPROXIMATE TRANSVERSE AXIS OF THE FLOWLINES SHALL NOT BE LESS THAN A/7 INCHES FROM THE BEARING SURFACE OF THE BOLT WHERE "A" IS THE NOMINAL DIAMETER OF THE SHANK AFTER HEADING.
5. (4) FATIGUE TESTING SHALL BE IN ACCORDANCE WITH NAS1069 AT THE LOAD LEVELS SPECIFIED HEREIN. THE LIFE REQUIREMENTS SHALL BE AS SPECIFIED IN SPEC MIL-B-7338.

LIST OF CURRENT SHEETS

NO.	REV.
1	4
2	2

CUSTODIAN: NATIONAL AEROSPACE STANDARDS COMMITTEE

NAS-624 through NAS-644 bolt chart.

HEAD MARKING	DESIGNATION AND STRENGTH	NOTES
	LOW STRENGTH BOLT OF UNCERTAIN MECHANICAL PROPERTIES	NOT TO BE USED FOR STRUCTURAL APPLICATIONS
	SAE GRADE 5 UTS 120,000 PSI	STANDARD MEDIUM STRENGTH INDUSTRIAL AND AUTOMOTIVE TENSION BOLT. NOW WIDELY COUNTERFEITED AND THEREFORE SUSPECT
	ASTM 325 UTS 120,000 PSI	IDENTICAL TO SAE GRADE 5 NOW WIDELY COUNTERFEITED AND THEREFORE SUSPECT
	SAE GRADE 8 UTS 150,000 PSI	STANDARD INDUSTRIAL AND AUTOMOTIVE HEAT TREATED HIGH STRENGTH TENSION BOLT NOW WIDELY COUNTERFEITED AND THEREFORE SUSPECT
	BOLT, MACHINE, AIRCRAFT AN73 THRU AN71 UTS 125,000 PSI	AVAILABLE IN BOTH UNF AND UNC THREADS BOLT HEAD IS HIGHER AN3 THRU AN20 SERIES BUT BOLTS ARE INTERCHANGEABLE
	BOLT, HEX HEAD, CLOSE TOLERANCE NAS 1303 THROUGH NAS 1320 UTS 160,000 PSI	DIMENSIONALLY SIMILAR TO AN3 THRU AN20 BUT SHANKS ARE SLIGHTLY LARGER IN DIAMETER FOR CLOSER FIT MUCH STRONGER THAN AN3 THRU AN20
	BOLT, 0156 OVERSIZE SHANK, CLOSE TOLERANCE NAS2903 THRU NAS2920 - UTS 160 000 PSI	SPECIAL PURPOSE OVERSIZE REPAIR BOLT WILL NOT FIT IN STANDARD HOLES IDENTICAL TO NAS3003 THRU NAS3020 IDENTIFIED BY LETTER "E" OFTEN FOUND SURPLUS
	BOLT, HEX HEAD, CLOSE TOLERANCE SHORT THREAD NAS1103 THRU NAS1120 UTS 160,000 PSI	CLOSE TOLERANCE SHEAR BOLT THIN HEAD AND SHORT THREAD TO BE USED ONLY IN DOUBLE SHEAR APPLICATIONS WEIGHT SAVER
	BOLT, SHEAR, CLOSE TOLERANCE NAS 464-3 THRU NAS 464-12 UTS 160,000 PSI	SIMILAR TO AND INTERCHAGEABLE WITH NAS 1103 THRU NAS 1120
	SINGLE RAISED OR RECESSED DASH DESIGNATES A CORROSION RESISTANT STEEL BOLT	TO BE USED ONLY FOR CORROSION RESISTANCE. STRENGTH, PARTICULARLY AT ELEVATED TEMPERATURES, IS LOW. LETTERS RM ARE AN OPTIONAL MANUFACTURER'S IDENTIFICATION
	TWO RAISED OR RECESSED DASHES DESIGNATE AN ALUMINUM ALLOY BOLT	LIGHTWEIGHT BUT LOW STRENGTH SPECIAL PURPOSE BOLT. DO NOT USE STANDARD TORQUE TABLES
	 NAS 147	BOLT, INTERNAL WRENCHING STEEL NAS144 THRU NAS 158 UTS 160 000 PSI AEROSPACE QUALITY INTERNAL WRECHING BOLT. INTERNAL HEX MAY STRIP WITH REPEATED USE.
	 NAS 624	BOLT, 12 POINT, EXTERNAL WRENCHING. NAS 624 THRU NAS 644 UTS 180,000 PSI MOST COMMON OF THE FAMILY OF NAS "SUPER BOLTS"

Bolt head identifying marks for SAE and aerospace bolts.

stocks at least the AN-3 through AN-20 stuff, and a little sniveling may get you at least a justifiable price.

The surplus market is hurting, but it is not yet dead. Both Earl's Performance Products and California Aero Supply in Los Angeles still have limited and unpredictable stocks of new surplus fasteners. Boeing surplus in Seattle is a virtual candy store. California Aero still sells surplus steel AN fasteners for $1.25 per pound (as of March 1989), but you have to do a lot of digging. There are certainly other surplus houses throughout the country, I just don't know about them.

You do not want to know about used surplus fasteners of any type—at any price. There is no way of knowing what the history of a used part has been, and you certainly do not want to find out the hard way that a bolt was near the end of its fatigue life when you bought it. Cleaning and replating a bolt does absolutely nothing for any aspect of its strength.

Going the surplus route means learning to identify and to measure the parts yourself. It also means doing a lot of looking, which can be rewarding if you have the time and enjoy browsing through interesting bits and pieces. Take a list of what you need by diameter, thread pitch, grip length and head configuration. Bring a micrometer caliper and be prepared to sort through a lot of boxes and bins. At the very least you will get inside some interesting places and meet some interesting people. You will also buy a collection of fascinating gadgets that you didn't know you needed but will come in handy if you never use them. Buy enough so that you don't have to go back very often. Carefully measure all surplus bolt diameters; many special-purpose oversized bolts get surplused. These bolts look absolutely standard to the naked eye and are useless.

Bolt Talk Four: Installation

If a bolt does not want to go into its hole there is usually a perfectly reasonable explanation. It is unlikely that a large hammer is going to help—and serious harm may result from its use. The cause is almost always misalignment of one or more parts. A tapered alignment pin and a little patience will usually solve the problem. In extreme cases, I use a dummy bolt with a gentle taper at the point. The dummy bolt is tapped and turned (not beaten) through from the backside to align the parts. The real bolt is then tapped through from the front side. This procedure leaves everything in its rightful place and the dummy bolt on the floor. I carry a whole bunch of dummy bolts around with me. If space does not allow this procedure and time is short, grind a 45 degree flat on one side of the point of the real bolt and tap/turn it through. The flat will cut the nylon locking ring on the nut, so use Loctite.

Mechanical design of the bolted joint

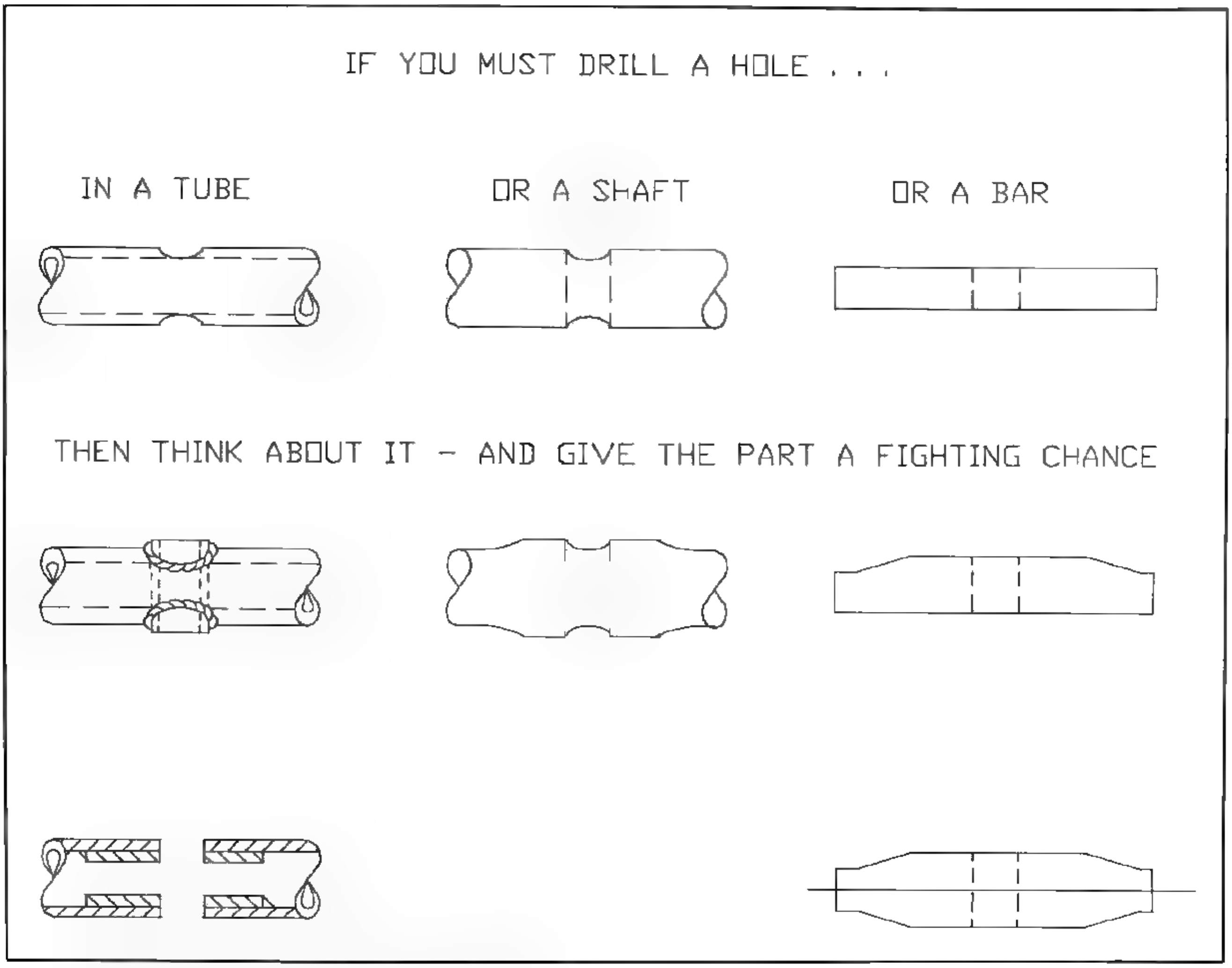
As used by the racer, the bolted joint is liable to have several strikes against it going in.

The engineer's rule is that God intended for bolts in tension to clamp surfaces together into rigid joints (not to be confused with rigid structures). He did not intend for bolts to be used in flexible or even partially flexible joints. The reason is that parts joined in a flexible manner, when loaded, will move in relation to each other—either in the plane of the bolt axis or perpendicular to it. Either way, the relative motion of the clamped surfaces will produce a stress in the bolt in addition to that foreseen by the designer of the joint. When bolting parts together, make sure that you have enough flange thickness to achieve rigidity, even if you have to add material that serves no other purpose. And never depend on bolts to locate the parts. Always bear in mind that clamping is the function of bolts and that location is the function of dowels. This is particularly true with respect to flywheels and ring gears, two areas in which designers are often remiss and, not by coincidence, two areas in which bolt failures are pretty common.

Bolts require holes; holes weaken whatever they are drilled in. Before drilling a hole in anything, think about what you are about to do. The man who designed the part that you are about to attack may very well not have meant for a hole to be drilled in it. Give the part a chance.

Rivets expand to fill their holes; bolts cannot. If we are going to be able to assemble (let alone disassemble) a bolted joint, the hole(s) must be larger than the bolt diameter. This makes the bolt hole, to some extent, an unfilled hole stress raiser. Bolt holes must be deburred—and peening them cannot hurt a thing.

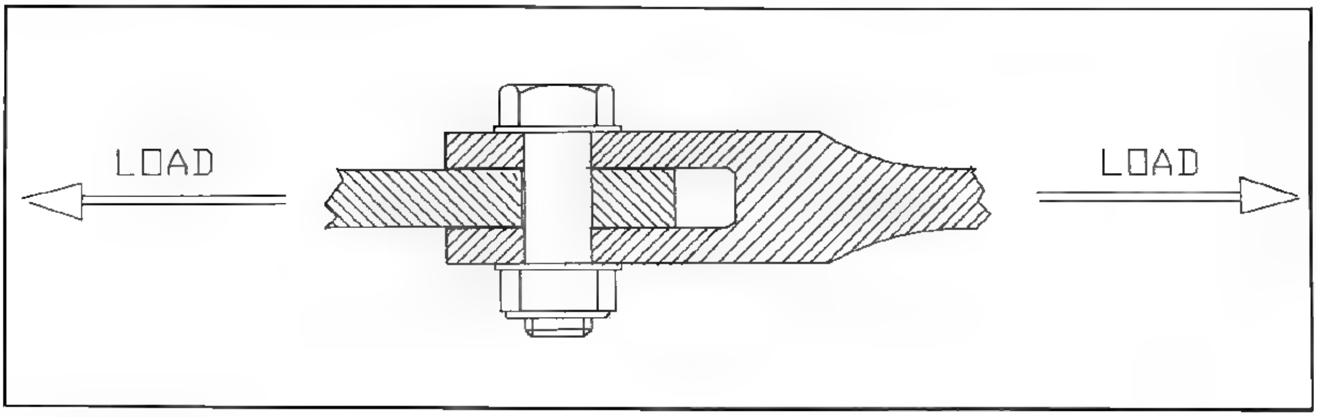
Since threads are by definition exceedingly efficient stress raisers, it behooves us to give the bolt manufacturers a bit of a hand in the avoidance of fatigue failure. To this end, follow the dictums of this book with regard to bolts and realize that highly stressed threaded fasteners are not reuseable.



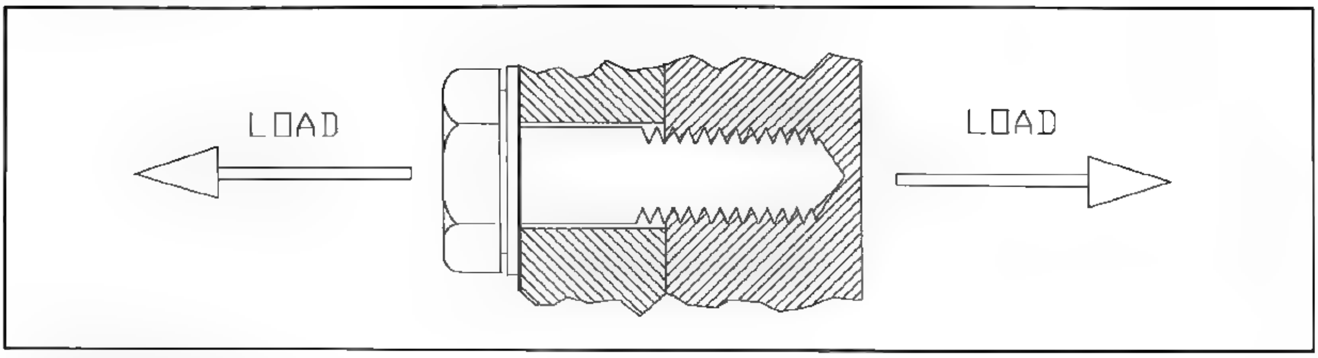
Give the part a chance—options in the design of holes.

Double shear joint

There are only two ways in which a bolt can be loaded: tension and shear. Of course, a bolt can be loaded in both tension and shear. If a bolt that is loaded in shear is supported on one side of the load only, the bolt is said to be installed in single shear. If the bolt is supported on both sides of the load, it's installed in double shear. Obviously the double shear mount is both stronger and more stable than the single shear mount. In fact, the single shear mount is a crime against nature and a perversion of the bad engineer.

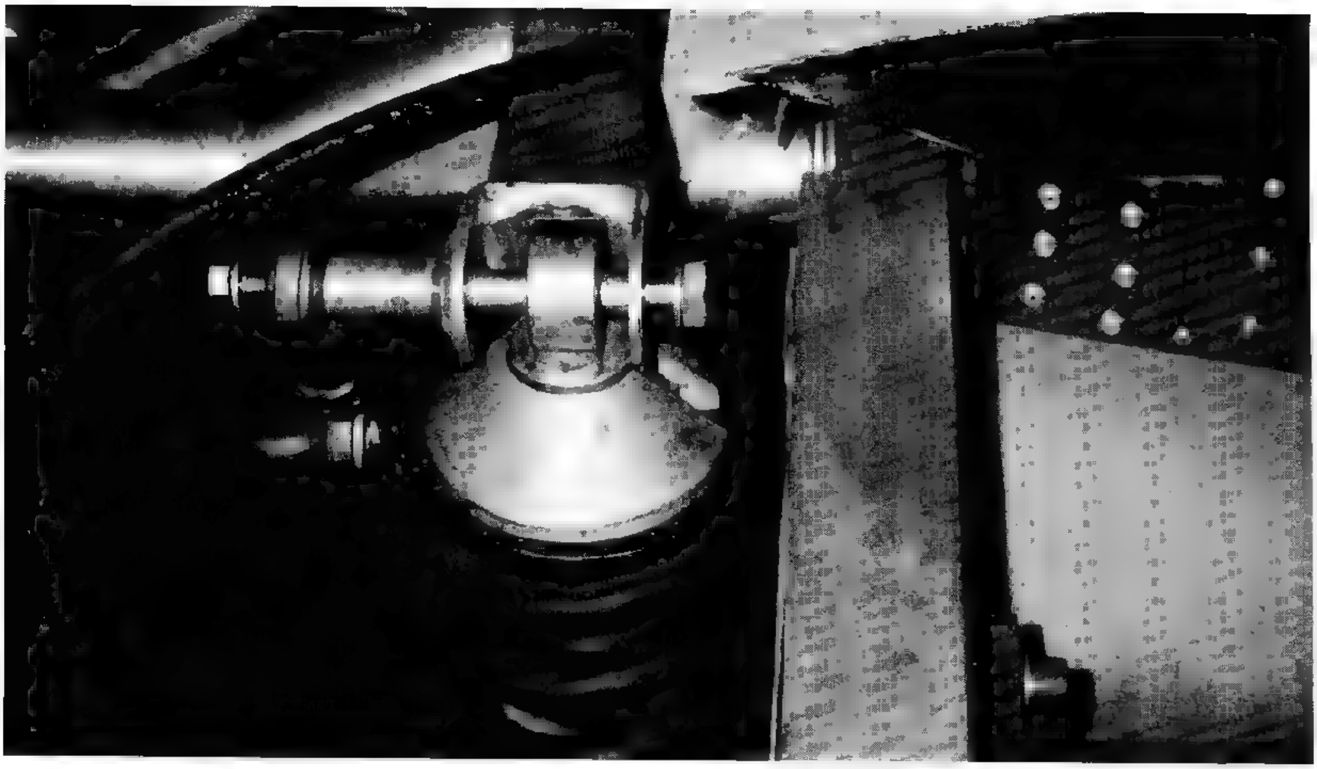


A bolt loaded in shear.

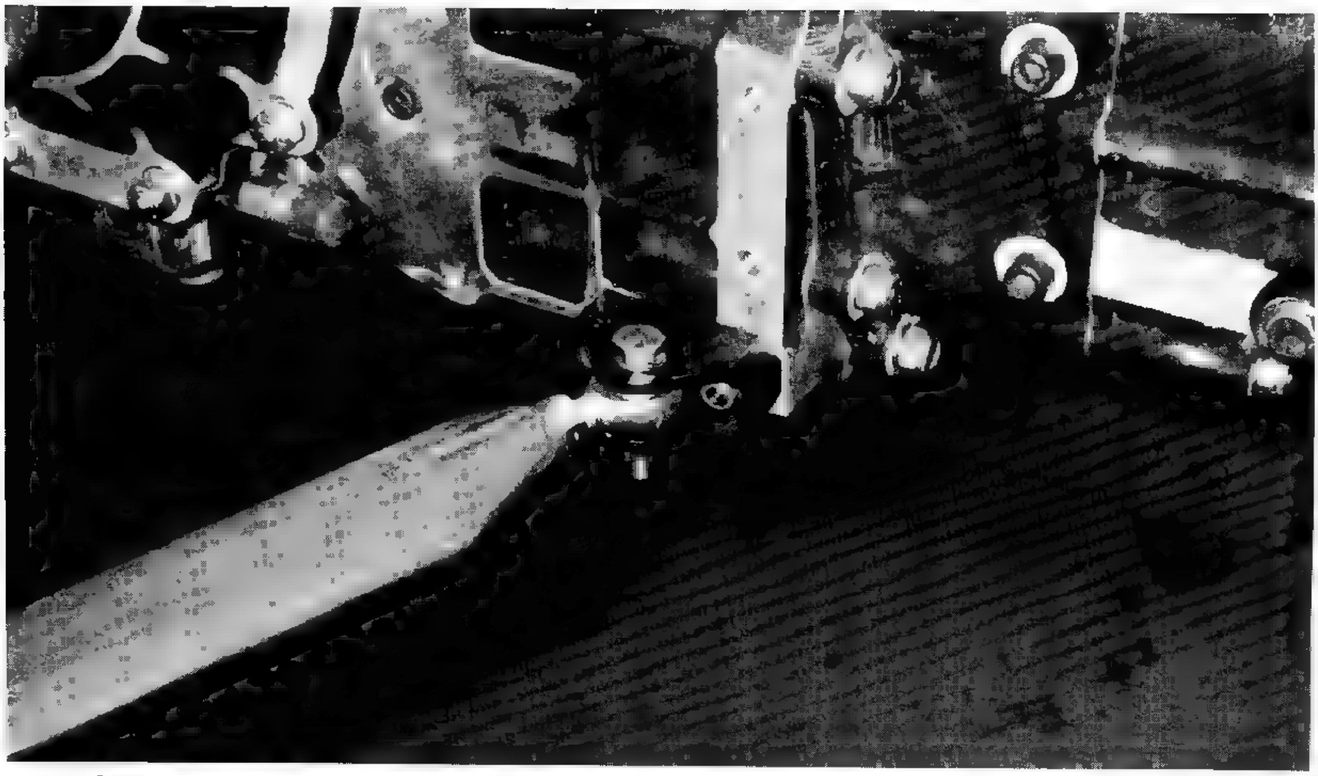


A bolt loaded in tension.

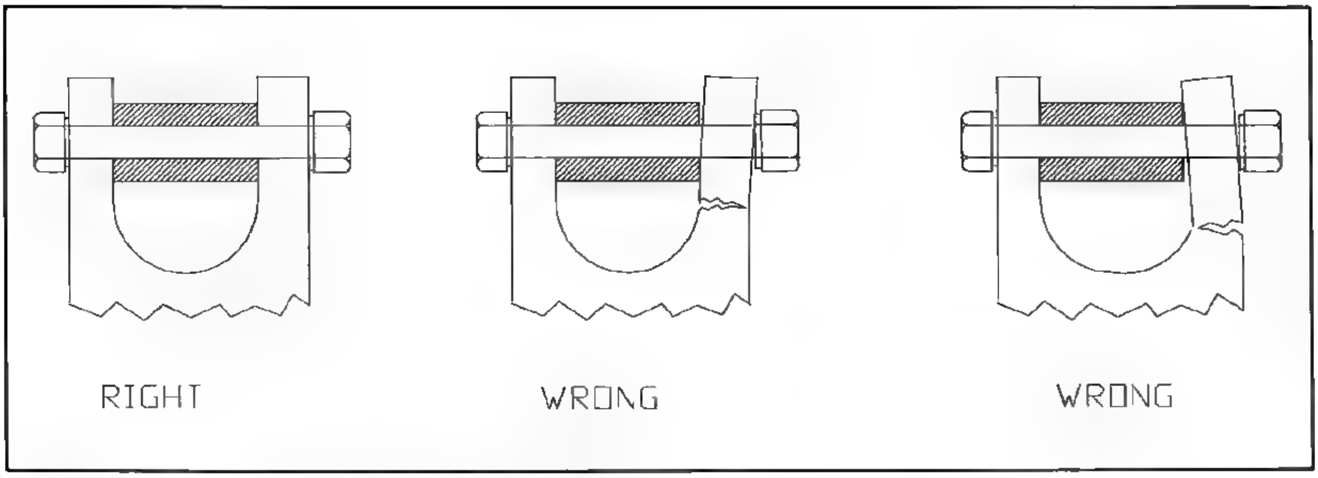
Bolt loaded in tension: Cosworth DFX connecting rod and rod bolt. Roy Kiesling



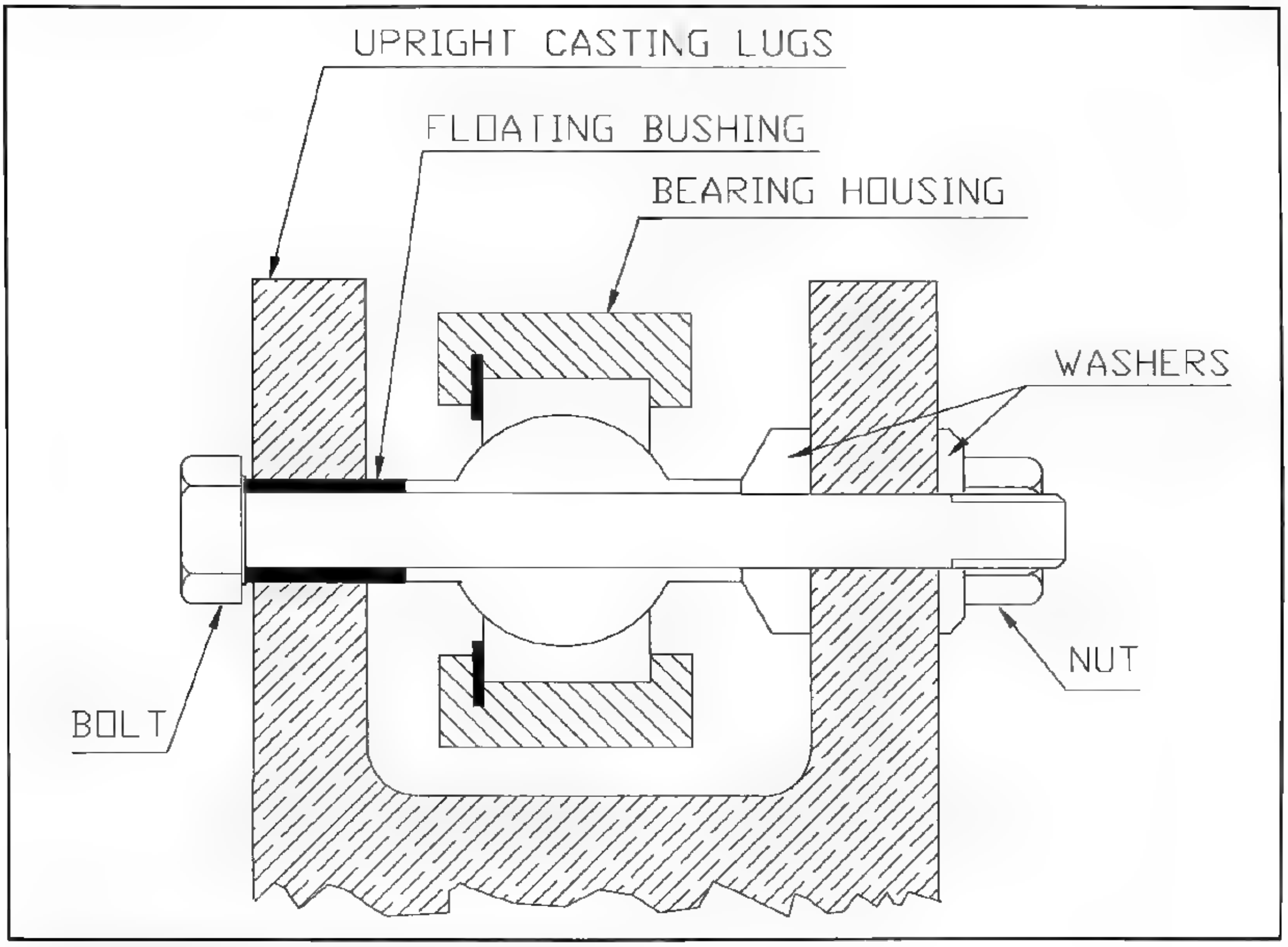
Bolts loaded in single and double shear.



A typical race car double shear mount used to transfer a tension/compression load from an articulated suspension member to major structure.



Double shear lugs unintentionally loaded in bending by poor mechanical fits.



A floating bushing installed in a double shear lug to both ensure a proper fit and to minimize crash damage to the lug.

Most of our critical chassis bolt applications are designed so that the bolt is installed in double shear. This is good. It is not, however, a panacea. The usual idea of our double shear joint is to transfer a load from an articulated member to main structure. The load will inevitably be either a tension or a compression load. It will be transferred from the articulated member by converting it into a bending load in the bolt. The bolt then transfers the load to the lugs or ears that form the double shear supports. From the lugs, the load flows into the structure to which the lugs are fixed—often by more bolts.

While the double shear joint forms an inherently stable system for load carrying and transfer, the design and fabrication of a proper double shear joint requires some thought. The first rule is that the bolt must be a close fit in the hole through each of the lugs. A bolt that is a loose fit in its hole is subject to both relative movement and high bending stress. It is therefore prone to premature failure from fatigue. In addition, the bolt will also introduce high bending stresses at the corners of the lugs—not good. Of course a loose bolt in any application will allow the clamped member to move under load, thus introducing unplanned relative motion between members into the equations of both vehicle dynamics and fatigue life. None of the above is desirable, and all will inevitably result from shear bolts that are a loose fit in their supports.

The second rule is that the holes in the opposing lugs must be aligned. If they are not, either the bolt cannot be installed, or beating it into place is going to generate still more unplanned-for stresses and attendant early failure.

The third rule is that the clamped member must be an exact fit in the space between the lugs. What we do not want is to allow the clamping force of the shear bolt to load the lugs in bending. The result will be a premature fatigue failure—of the lugs, not of the bolt. This is more critical when the lugs in question are made from relatively brittle cast aluminum, magnesium or iron than when they are made of steel.

One of the clever features of the Anson Formula Three and Super Vee racing cars was Gary Anderson's solution to this problem. The use of a floating bushing not only takes the criticalness out of the fit, it greatly reduces the probability of breaking the ears in a crash. David Bruns uses the same concept at the attachment of the lower rear control arm to the Swift bellhousing.

The design of the lugs in the double shear joint is at least as important as the strength of the bolt—perhaps more so. We racers typically do a miserable job in this area. A thin lug (like a piece of 16 gauge mild steel with a hole drilled in it) is dead easy to make and won't introduce bending loads into the bolt simply because it will not be strong enough to do so.

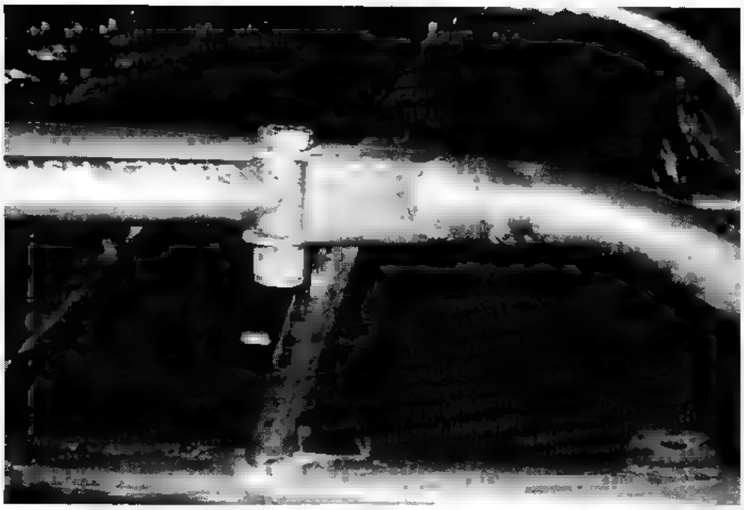
It will also provide insufficient bearing area for the bolt. This means that the stress at the edge of even a properly sized hole will be infinitely greater than the stress in the rest of the lug. The hole will therefore have a nasty tendency to elongate under load. On the other hand, an overly thick lug will be unnecessarily heavy and difficult to make. In extreme cases it can also transfer the bending stress caused by the load from the lugs into the bolt. The bolt will then bend and again the edges of the lug will be overstressed. Shucks!

The fastener industry recommends that each lug have a minimum thickness of about ⅓ the bolt diameter, when the strength of the lug material is about equal to the strength of the bolt in shear. Maximum recommended thickness, regardless of material, is about equal to the bolt diameter. Aerospace engineers spend a lot of time worrying about this sort of thing. I do not, except in off-road applications. I tend to use bolts that are a lot stronger than they need to be. For sheet-metal lugs I calculate the tear-out load for the lug, and multiply it by *four.* I then use the next greater thickness of standard sheet metal for the lug, stitch or spot weld the next undersized washer to the outside of each lug, drill an undersized hole and ream the assembly to size. This results in a lug that will never fail under load, doesn't transfer much bending load to the bolt, has sufficient bolt bearing area so that the holes will not elongate in service, is not excessively heavy and is easy to make. It works out that I make most of my steel lugs from 0.095 in. sheet steel. I follow a similar procedure for machined lugs except that I don't do any welding. Refer to the list for bearing area information for different sheet thicknesses and bolt diameters.

Double shear bolt as a trunnion

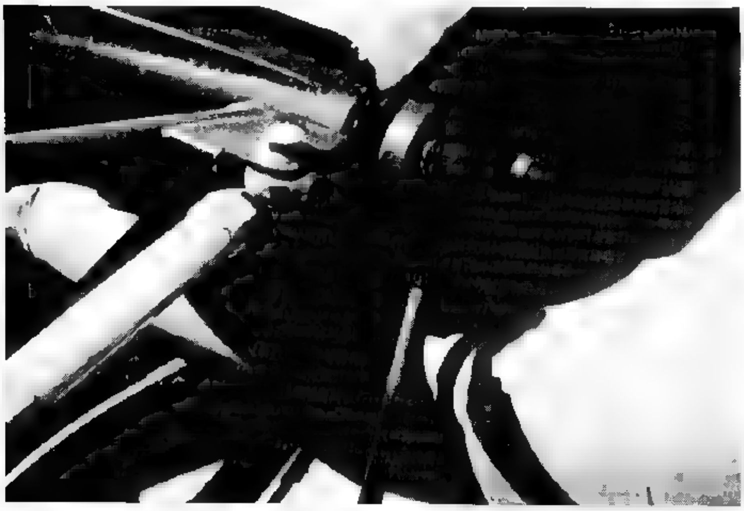
God did not really intend that bolts be used as axles or trunnions. Aircraft designers found this to be inconvenient and developed a whole series of airframe bolts that are perfectly acceptable for use as trunnions in double shear mounts for bearings or

A typical machined double shear lug.



Sheet-metal double shear lug welded to tubing.

Nicely crafted double shear lugs.



Sheet-metal double shear lug.

bushings—but only when the bolt itself is well supported in double shear and when the bearing is not going to rotate on the bolt. In no case should any rotating or oscillating member be allowed to turn directly on a bolt. The proper way, indeed, the only safe way, is to use the bolt to clamp a bearing between supporting members. Note that nothing in this assembly rotates on, or has any relative motion with respect to, the bolt. The mounting itself must be acceptable with respect to bearing area, ultimate strength and fatigue life.

We racers use bolts as trunnions for all kinds of suspension bearings and/or bushings. This is OK as long as we exercise the same care as the aircraft people do: use the right type of bolt, and make sure that we have enough bearing area, both in the lugs and in the articulated member. For the articulated member, this last provision is almost automatically taken care of by the bore diameter of the bearing/bushing involved. But this means nothing to the mounting lugs. One of the less well known features of the SAE-graded bolts is that SAE bolt shanks are 0.002 to 0.004 in. undersized compared to the AN/MS items. Standard AN and aerospace bolts are 0.001 to 0.005 in. undersized to start with. This means that the aircraft bolt is a considerably better fit in the rod end bearing, trunnion, yoke or whatever else we may be holding in shear. If a better fit is needed, we have two choices. First, we can use a close tolerance bolt such as the AN-173 through AN-186 series or any of the NAS bolts, which are typically 0.0008 to 0.0015 in. undersize. Or we can machine the almost inevitable top hat spacers that will fit inside the bearing bore to be a precise fit on a standard bolt.

When designing shear lugs, pay attention to the forthcoming section on sculptured structure and make damned sure that the bolt is a good fit in a reamed hole in the lug.

Residual bolt stress and the shear joint

At first glance it would seem that installed bolt stress is important only when the joint is loaded in tension. Wrong again! Since most machinery and vehicle joints are subject to cyclic or reversing loads, relative movement between the clamped parts of the assembly becomes a factor in both the performance and the fatigue life of the joint. Since the bolt is not normally a press fit in its hole(s), and since most installations are not piloted or doweled, this relative movement (and consequent fretting and loosening) is resisted both by the residual stress in the bolt (clamping force) and by the friction between the assembled parts.

The frictional force is more important than it might appear. In fact, the most effective method of increasing the fatigue life of a shear joint is to improve the surface finish of the clamped parts (this is why a thin layer of silicone seal between fretting parts is a valid temporary fix). Anyway, just as in the tension joint, the more residual stress in

Bearing strength of 4130–N steel sheets with bolts and rivets

Rivet bolt diameter (in.)	⅛	5/32	3/16	¼	5/16	⅜	7/16	½	⅝
Sheet thickness	Bearing strength of sheet (lb.)								
0.035 in.	438	547	656						
0.049	612	766	919	1,225					
0.065	812	1,016	1,219	1,625	2,030				
0.072	900	1,125	1,350	1,800	2,250	2,700			
0.083	1,038	1,298	1,556	2,075	2,594	3,112			
0.095	1,188	1,484	1,781	2,375	2,970	3,560	4,750		
0.120	1,500	1,875	2,250	3,000	3,750	4,500	6,000	7,500	
0.188	2,344	2,930	3,510	4,680	5,860	7,030	9,375	11,720	
0.250	3,125	3,900	4,680	6,250	7,800	9,375	12,500	1,5625	18,750

Bearing strength of various thicknesses of 4130–N steel sheets with different diameters of bolts or rivets.

Bearing strengths (lb.) of steel sheets on bolts and pins

Source: ANC–5

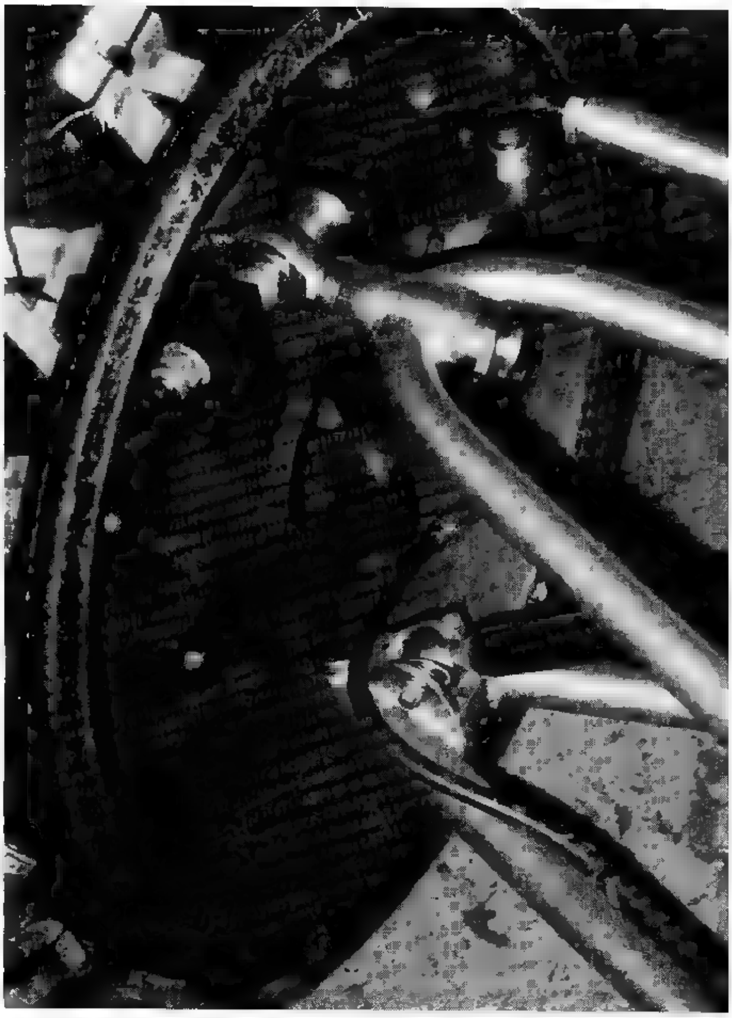
Bolt diam. (in.)	3/16	¼	5/16	⅜	½
Sheet thickness					
3/16 in.	3,515	4,688	5,860	7,030	9,375
¼	4,688	6,250	7,810	9,375	12,500

Bearing strength (lb.) of steel sheets on bolts and pins.

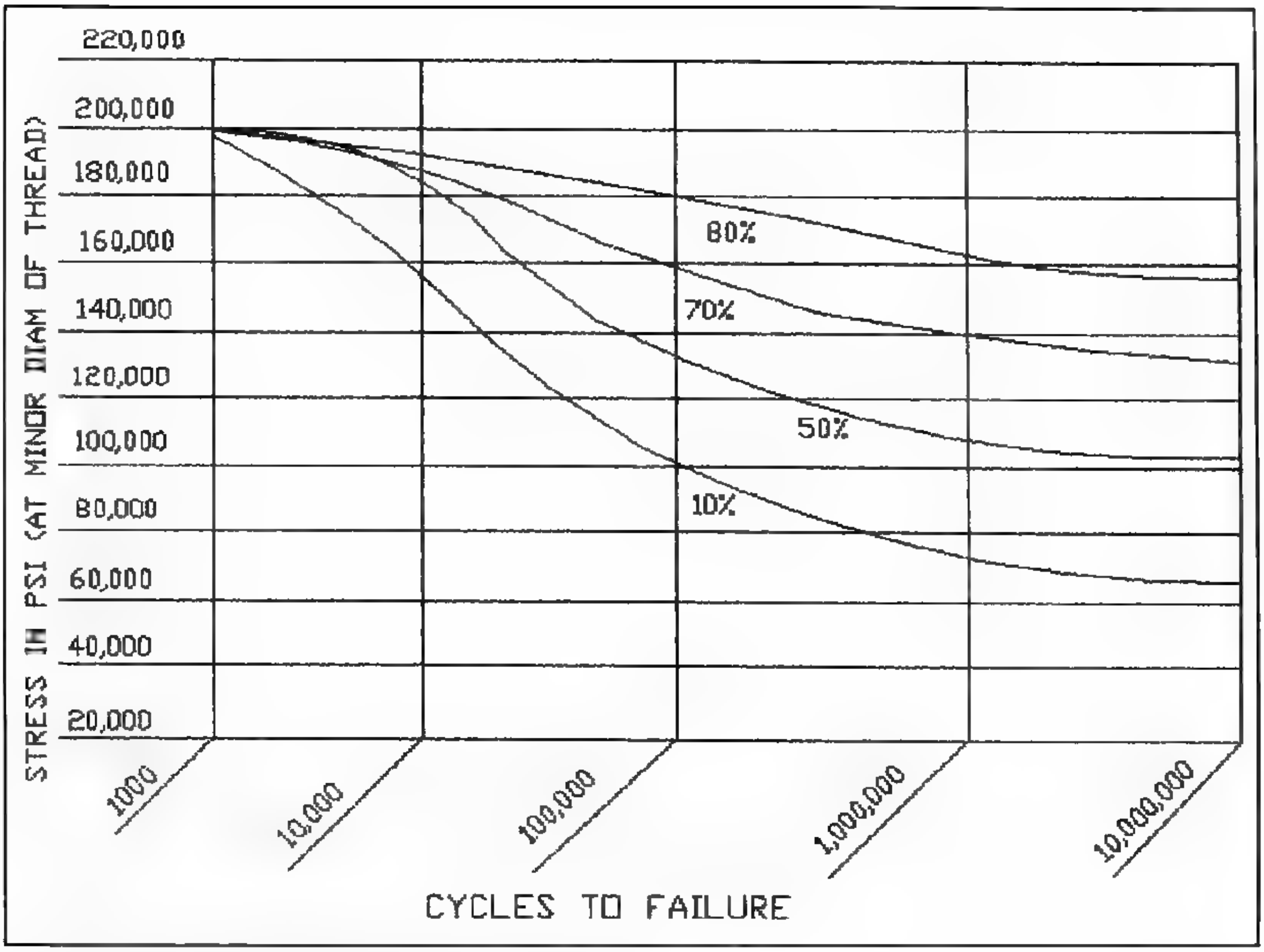
the bolt, the more clamping force. In the shear joint, the more clamping force, the more frictional force in the shear planes of the joint and the longer the assembly will last under cyclic stress.

When a bolt is installed in tension, the entire shank is stressed in tension. One of the least appreciated factors in bolt use is the fact that the threaded portion of a bolt shank should never be loaded in shear. There are several reasons for this. First, the major diameter of the thread is slightly smaller than the diameter of the unthreaded portion of the bolt shank. If the threaded portion of a bolt is located within a shear lug, there is no way the bolt can be a good fit in the hole. While a loose-fitting tension bolt shank is a necessity, a loose-fitting shear bolt shank is a crime against nature.

Second, the threaded portion of the bolt has much less surface area than the unthreaded portion. It therefore offers significantly less bearing



Properly designed double shear trunnion mount.



The effect of various levels of residual stress or preload on the fatigue limit of a bolt in tension.

area to the lug. This reduces both the load carrying capacity and the fatigue resistance of the assembly.

Third, the load bearing cross section of the threaded portion of the bolt is less than that of the unthreaded portion. Since it is the cross section that must resist both shear and tension loads, the threaded portion of the bolt shank is weaker and less able to accept loads and to resist fatigue than the unthreaded portion. It makes no sense at all to stess the weakest part of the bolt in shear when you don't have to.

Fourth, if there is any relative motion at all between the thread and the lug, the thread will act as a low-speed file and do unspeakable things to the hole.

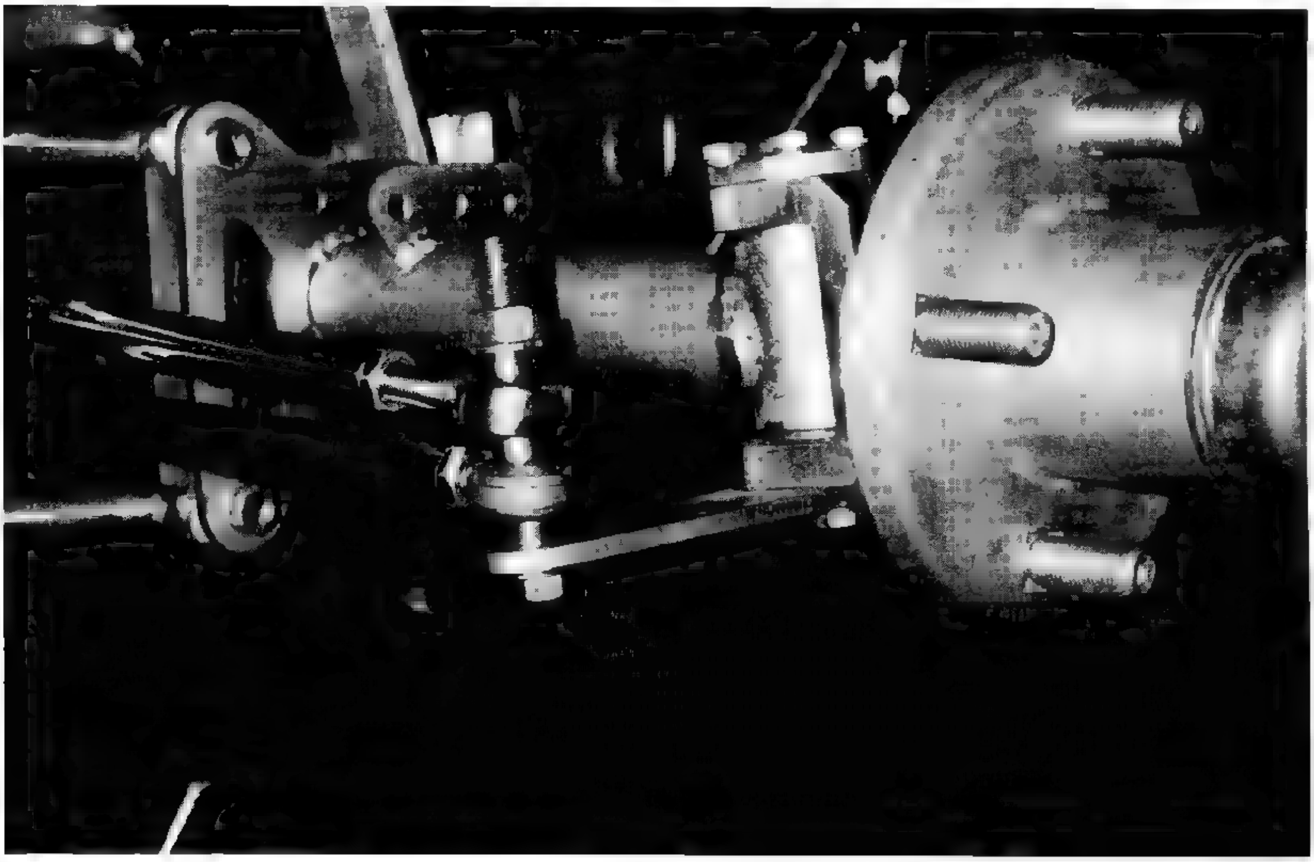
The solution is simple—use a bolt with the correct grip length. If the right bolt is not available, use the next available longer bolt and more washers. *Never* die cut threads onto a bolt in order to shorten it. The thread die leaves jagged tears at the thread root and at the run-out to the unthreaded shank. The die-cut thread is one of the all-time great stress raisers, and the lathe-cut thread is only slightly better. Lathe-cut threads are marginally acceptable when the bolt is loaded in double shear. They are not acceptable at all in tension applications, and are deadly in single shear.

This has probably been more detail than I can justify. I have included it, not to make you paranoid, but to point out that *every* detail of the racing car (or the aircraft, or the high-performance anything) needs to be thought about before it is put into metal. There is a popular attitude that, finding weak points is what vehicle testing is for. This attitude is just plain irresponsible—and stupid. Always remember that when one part of the car fails, the ensuing gyrations and/or the sudden final stop are liable to damage a lot more parts, each of which will require both time and money to repair or replace. Besides, it is embarrassing for the designer, and someone is liable to get hurt. It is a hell of a lot cheaper to make the part right the first time and to devote expensive testing time to making the car go fast.

Bolts in single shear

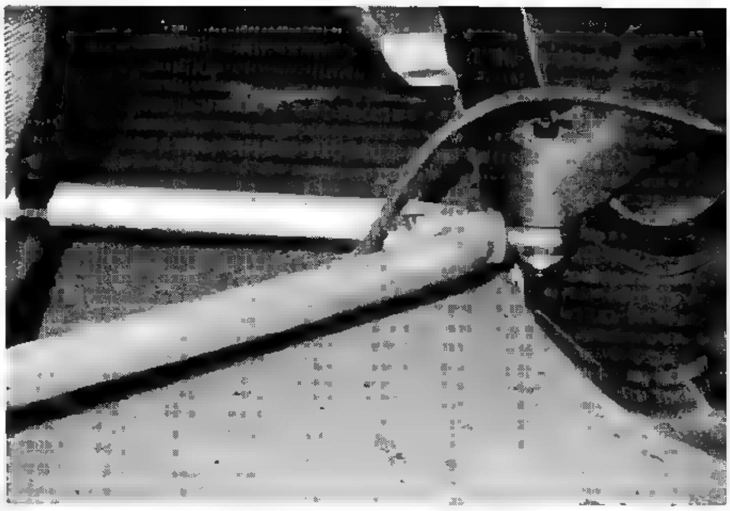
Some years ago, most of the English racing car designers seemed to specialize in installing suspension bolts in single shear. They got away with it—most of the time—by using bolts about twice as large in diameter as were actually required. The Brits know better now. Regretably their place, in this respect at least, has been taken over by the designers of off-road racing cars, sprint cars, dragsters and stock cars. I have only three words of advice when it comes to single shear applications: Don't do it!

There is only one option offered and both alternatives are bad. You can either place the run-



A single shear bolt mount—about as bad as it gets.

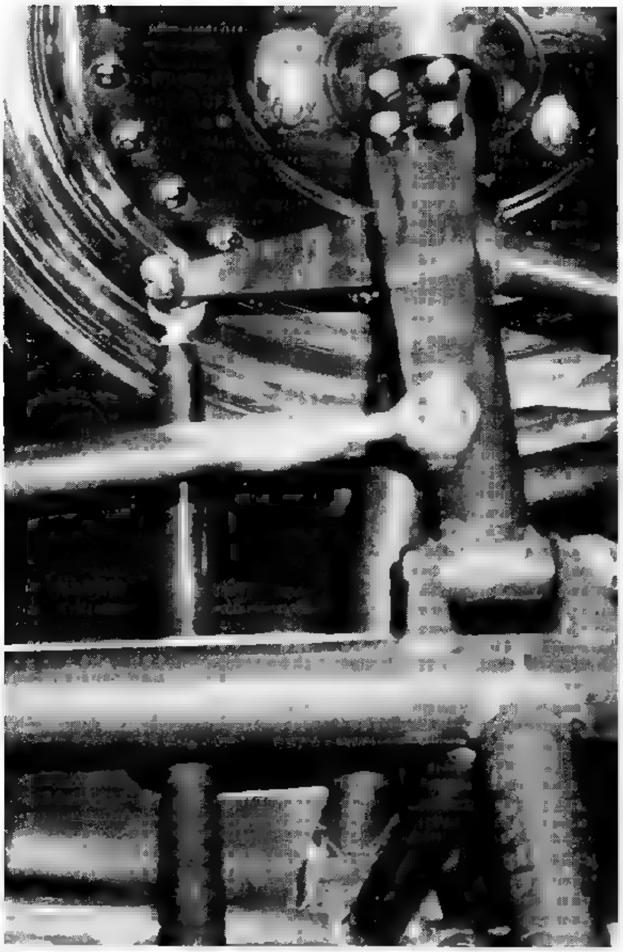
Single shear lug installation.



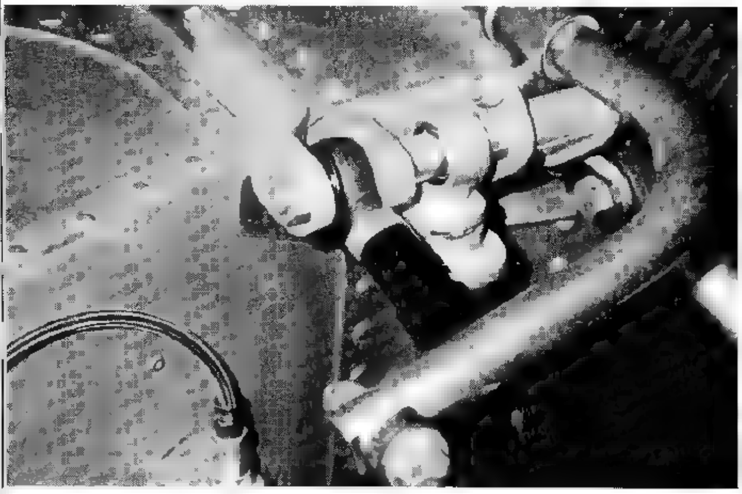
Single shear A-arm mount.

out thread of the bolt at the joint interface or you can place it inside the bearing. In the first case the bending load and attendant shear stress are placed at the run-out thread of the bolt. That is almost like a dotted line that says, "tear here." In the second case the run-out thread is hidden inside the bearing so as to guarantee a loose fit between bolt and bearing.

If you must load a bolt in single shear, use a very wide lug and hide the run-out thread inside the lug, not in the articulated member. If you have to use a stud instead of a bolt to accomplish this end, do it! The bottom line is to keep the run-out thread away from the joint interface and to maintain sufficient bearing area—combined with a reasonable radial fit in both parts of the joint. A better solution is to modify the application so that the bolt is loaded in double shear.



Single shear applications.



A neat approach to converting a single shear mount to a double shear mount.

Single shear to double shear conversions with addition of plate.

Single shear to double shear conversions with internal-wrenching bolts.

Sizing the bolt hole

Many years ago the aerospace people worked out the specifications for bolt holes according to type of loading and severity of service. As an example, in 1950 the Lockheed Aircraft Company came up with the following classes of fit:

Extra loose drilled and loose drilled holes: These holes are specified when bolts are to be loaded in shear or in combined tension and shear, and are to be installed with high margins of safety where interchangeability and speed of assembly are of prime importance. As an example, while a loose drilled hole should not be used in a sheet on which plate nuts are mounted, it should be used in the mating part where possible.

When the bolt is to be loaded only in tension, the loose drilled hole is the standard hole and is used whenever a closer fit is not required for alignment of parts. Since bolts are meant to be clamps and not locating devices, a closer fit for a tension bolt should, theoretically, never be required.

Clearance drilled holes: For installations where the bolt is loaded in shear or in combined tension and shear, the clearance drilled hole is used where there is an oversufficiency of bearing area and where the joint will not be subjected to shock, vibration or frequent reversal of load. For tension applications, the clearance drilled hole should be used only when a loose drilled hole will not provide proper alignment.

Close drilled holes: When the bolts are to be loaded in shear or in combined tension and shear, this fit is used with relatively low margins of safety in bearing area. When four or more bolts are employed, close drilled holes are satisfactory for use where loads are applied with shock or vibration or when frequent reversals of load are expected. When less than four bolts are used, the close drilled hole should be restricted from such service. In tension applications this fit should be used only when required for alignment—which should be never.

Reamed holes: This fit should be used only with ground close tolerance bolts loaded in shear or in combined shear and tension. It should be used in all connections using less than four bolts

Single shear to double shear conversion with use of plate.

Application of bolt holes to design

Source: Lockheed Aircraft Company, copyright 1949

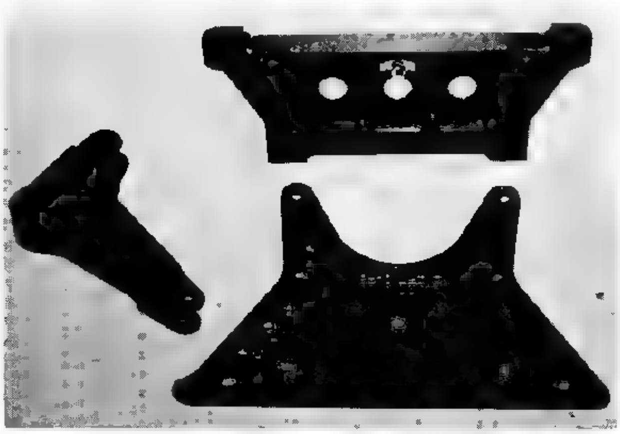
Bolt or pin size	AN Bolt diam. (in.)	Extra loose drilled (in.)	Loose drilled (in.)	Clearance drilled (in.)	Close drilled (in.)
#10	.186/.189	.218/.225	.203/.213	.196/.203	.191/.198
¼	.246/.249	.287/.295	.269/.277	.258/.266	.249/.257
5/16	.309/.312	.356/.364	.336/.344	.325/.333	.313/.321
⅜	.371/.374	.421/.428	.401/.409	.388/.396	.375/.383
7/16	.433/.437	.483/.490	.468/.475	.452/.459	.437/.444
½	.495/.499	.546/.553	.530/.537	.515/.522	.499/.5066

Lockheed Aircraft Company dimensions for standard holes.

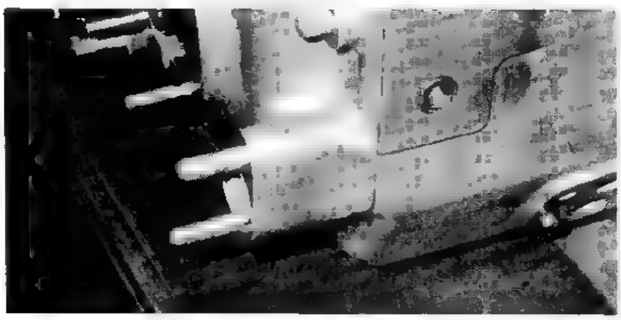
when loads will be applied with shock or vibration or when frequent reversals of stress are expected. The table shown here lists Lockheed's dimensions for the various types of standard holes.

The US Navy Bureau of Aeronautics uses a slightly different method for bolts loaded in shear:

Fit	Tolerance	Use
Class 1	+/- 0.001 in.	To be used when one or two bolts are subjected to a reversal of loads in a critical joint assembly. Holes are reamed on assembly.
Class 2	+/- 0.002 in.	To be used where there are more than two through primary structures, where vibration and reversal of loads are expected or where joint rigidity is required. Holes are reamed on assembly.
Class 3	+/- 0.005 to +/- 0.010 in.	To be used where there is a large number of bolts not subjected to load reversals.



Sculptured structure.



Relieving the unthreaded shank and the run-out thread of highly stressed bolts.

What all of this is telling us, of course, is that the racing car requires reamed bolt holes for all shear applications. The only way I know of to arrive at acceptable bolt holes for shear applications is to drill the holes well undersize in a drill press, redrill 1/64 in. undersize and finish with a reamer.

Bolts in tension

While discussing shear applications, I stated that tension is different. We chassis and suspension people are so used to double shear bolt applications that we lose sight of the fact that, in the real world, most bolts are installed in tension. Let's look at the tension bolt for a moment.

We have been taught that the ideal length of engaged thread in a tension bolt (or stud) application is 1.5 times the bolt diameter. Being weight conscious, we racers tend to regard any more thread than that as dead weight. What we are ignoring here is the word engaged—and we can get ourselves into a lot of trouble by doing so. Referring to the illustration on page 89, we first see a tension bolt with 1.5d of engaged thread—the ideal setup, right? Wrong! What we have achieved here is the placement of the weakest part of the bolt—the root of the run-out thread—at the interface between the clamped parts. If the bolt is properly tightened, the bolt itself will be subjected to a preload that will stress the bolt to a level close to its yield point.

This is fine. Indeed it is the stress level at which the bolt was designed to operate. But any relative motion at all between the clamped parts will subject the bolt to an additional stress, either in tension or in shear. This additional load will constitute a design overload and the bolt will fail prematurely from fatigue—at the thread root. The situation is not helped by the fact that the installed tensile stress is not evenly distributed throughout the length of engaged thread but is largely concentrated in those threads closest to the joint interface. So placing the run-out thread near the interface is doubly inviting trouble.

The proper way to install a bolt that will be loaded in tension is also shown. The thread is extended into the hole in the top member so that the weak run-out thread root is removed from the danger zone of shear stress. Alternatively, the tapped hole in the lower member can be counterbored so that the unengaged thread can extend into the lower member. This has been Ford Motor Company's solution for cylinder head bolts for decades. It is all very well to say that the assembly should be designed so that there will be no relative motion between the clamped parts or that bolts should be used as clamping members only, not as locators—but we do not live, or work, in a perfect world.

The second thing that we are forgetting when we take the 1.5d length of engaged thread as gospel is the simple fact that the rule assumes that the

female thread is made from the same or at least similar material as the bolt. When we are talking about a weaker (softer) material, like an MS20074 bolt into an aluminum or magnesium casting, all of the other factors that we have just discussed remain the same but the recommended length of engaged thread doubles—to 3 diameters!

Sculptured structure

Sculptured structure is a relatively new term. In many cases it is not only possible, it is desirable, to thin down sections of machined components. There is no sense making the whole damned part thick and heavy if it doesn't have to be. On the other hand, local increases in section thickness are often necessary to provide adequate bearing area for through bolts or rivets.

This locally built-up type of design is termed sculptured structure. The drawings and photos shown here illustrate the concept. The aerospace people go so far as to chemically mill fuselage and wing skins so that the rivet lines are thicker than the skins themselves. Racers do it only on plate and bar stock. The tape-controlled universal milling machine makes this practical—even easy. It is also sound engineering, and the practice has a certain inherent elegance that I appreciate.

Why the threads are so long

In the worlds of race cars and aircraft, most bolt applications are loaded in double shear. In the *real* world, however, most bolt applications are loaded in tension.

When engaged in shortening the threads and dressing the ends of nonaircraft bolts so that you can use them on your race car, you may well wonder why the damned fools who designed the things made the threads too long to start with. As is usual in commercial engineering, there are perfectly good reasons behind this seeming idiocy. Most obviously, the use of a very long thread allows a reasonable stock of bolts to cover a wide range of applications.

Not so obvious, but a lot more important, is the fact that neither the manufacturer nor the seller has any idea what the bolt is going to be used for. While it is likely that few members of either of these groups have any sort of conscience, the SAE does—and since they are the ones who get to lay out the fastener specs, we get long threads. Any bolt that you buy in a hardware store or industrial supply house will, within the envelope of its material strength and detail design/fabrication, be safe when it is used in tension, regardless of the material.

Bolt Talk Five: Failure

So why do we experience bolt failures? There are several reasons. In no particular order they include:

First, underestimation of the load(s) to be experienced. The instant the load on a bolt exceeds the residual tension in the bolt, the excess load adds to the bolt stress and the bolt stretches. If the total load is sufficiently above that which the designer predicted, the result can be, and often is, instantaneous and catastropic failure. A lesser excess load will result in the more common premature failure from fatigue.

Second, insufficient tightening of the bolt resulting in insufficient residual stress.

Third, use of a bolt of inferior quality—either in strength or in resistance to fatigue—to that specified in the original design. This, of course, is what is going on in the military, the NRC and NASA with counterfeit bolts selected by the low bid process.

Fourth, improper design or assembly procedures, including but not limited to: Improper hole sizing or alignment; improper grip length of the bolt—for example, placing the last thread at the part interface; damage to threads, as in extending the length of thread with a die; damage to the shank, as in "the hacksaw slipped but it's just a little nick—it'll be OK!"; fatigue due to cyclical loading—but only if the loads or the number of cycles are in excess of what the designer had in mind; attempting to make a nonrigid or flexible joint live under cyclic loading; attempting to install either a bolt or a nut in such a way that the bearing face is not parallel to the work face, putting a bending load on the bolt and leading to instant failure under load; insufficient length of engaged thread.

Cyclic stress and fatigue

When you consider the connecting rod bolts in a racing engine, or the landing gear bolts in a carrier-based aircraft, it is pretty obvious that it is indeed possible to design a bolt installation that will live under repeated high levels of cyclic stress—but not forever. Highly stressed bolts should always be lifed in terms of hours of service (measuring stress levels and cycles in a dynamic installation is, as yet, impractical). We can demonstrate that, when the level of cyclic stress due to applied loads approaches the residual tensile stress in an installed bolt, failure from fatigue will shortly follow.

When this does occur, there are several alternative courses of action. There is the typical nonthinking approach: replace the broken bolts with identical new ones. This approach seldom produces acceptable results. You can also use either more or stronger bolts. Very often an open-minded re-examination of the installation will reveal conditions of loading that were overlooked in the original design. As an example, a highly regarded Formula One team recently suffered a series of testing failures in which the ring gear bolts in a new transaxle failed in tension. The design had been properly stressed in shear, but the designers had totally overlooked the enormous spreading loads placed on the differential case by the Salisbury-type

limited slip—and the ring gear bolts also held the case together . . .

In many cases, however, the solution is much more simple, if not so obvious. We can greatly increase the fatigue life of many bolted assemblies by the simple act of properly tightening the bolts. As an example, in laboratory tests an AN–6 bolt, tightened to 1,420 lb. of tension (29,000 psi residual stress) was subjected to a load of 9,000 lb. in alternating tension and compression. The bolt failed after 6,000 cycles. Identical bolts tightened to a tension of 8,420 lb. (80,000 psi residual stress) and cycled at the same 9,000 lb. load went 4,650,000 cycles before failure. Of course the bolts in question were not properly tightened to begin with, but surprisingly few are, largely because people do not understand how tight bolts should be.

What we actually want to do is tighten the bolt to the point where the residual stress in the bolt will always be greater than the stress produced by the applied load. In order to achieve assemblies of minimum size and weight (as in racing cars and aircraft), we tighten the bolts to just below their yield strength. This is a perfectly acceptable practice—if we have a foolproof method of measuring this point. Usually we do not, and that is where we get into trouble.

A great many capable people will tell you that you cannot tighten a bolt to this level because it will give up. What they are actually talking about is what happens when we exceed the yield strength of the bolt while tightening—the famous old, "Tighten it until it feels funny and then back it off a little." As a point of interest, in construction and in ship building, large bolts are very accurately prestressed by turning the fastener until contact is made with the work surface and then tightening it a specified number of flats. The specified number of flats is calculated to equal the strain produced by the desired residual tensile strength in the bolt. This method is only practical with large diameter bolts of one length, but it is accurate and damned near foolproof.

Very highly stressed bolts and studs

It does not require engineering genius to figure out that a chain will always break at its weakest link; the trick lies in figuring out which is the weakest link. In the case of the bolt loaded in tension there is no trick at all. The bolt will fail either at a stress raiser or at its smallest cross section. In a conventional bolt the smallest cross section is found in the threads. So are the stress raisers. This means that the unit stress will be higher in the threads than in the shank, as will be the strain. Furthermore, the stress is not evenly distributed along the length of the engaged threads, but is concentrated on the threads nearest the female thread work face. This is unfortunate for a couple of reasons:

First, the threads are necessarily full of stress raisers and so should be less severely stressed than the shank—which is relatively free of stress raisers.

Second, for maximum fatigue life we would really like both stress and strain to be evenly distributed throughout the length of the bolt, including the threads.

You may have noticed that on some very highly stressed bolts (and studs) the diameter of the shank has been reduced over a part of its length. The thinking here is that for maximum clamping force and maximum resistance to fatigue, what is needed is an even distribution of stress throughout the length of the fastener. But the weakest portion of the bolt is the minor diameter of the threaded portion. This means that, when the bolt is tensioned, stress will concentrate in this already weak area (which is also full of stress raisers) while the stronger shank will be relatively lightly stressed—a backwards state of affairs if ever one existed.

The solution is to reduce the diameter of the unthreaded shank to slightly less than the minor diameter of the thread. If the diameter change is properly radiused, the tension stress will then be evenly distributed throughout the length of the fastener. This will allow us to take maximum advantage of the strength of the alloy used and to achieve maximum resistance to fatigue. Selected areas of the unthreaded shank must be left at the standard diameter in order to locate the fastener in its hole. The drawings and photos shown here illustrate what I am talking about and also give the proper dimensions for the stress relief groove at the runout thread. They prove that the technique is no secret.

We racers normally see this sort of thing only on connecting rod and cylinder head bolts. Farmers and truckers see it all the time on farm equipment and diesels.

Bolt Talk Six: Joint rigidity

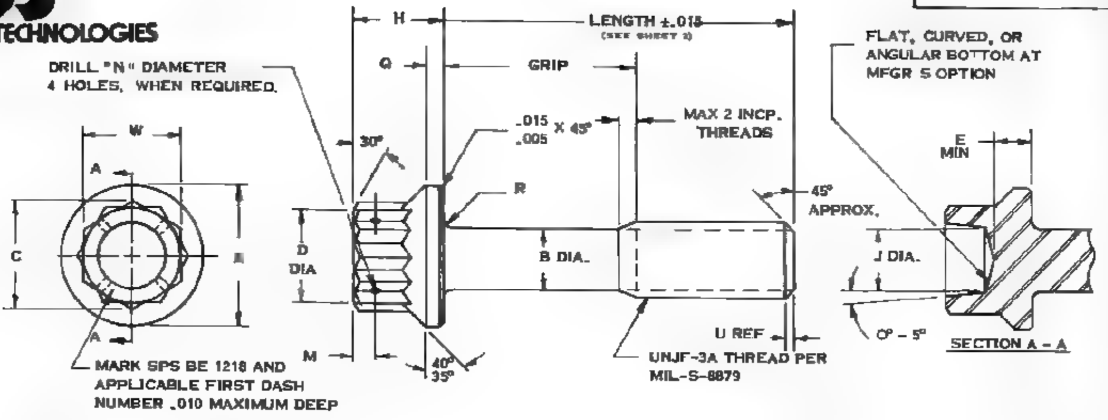
I have frequently mentioned the desirability of the rigid joint. This does not mean that the assembly has to be rigid. There is no such thing as a truly rigid structure, nor should there be. A rigid structure would not be able to give in response to an applied load and so spread the load throughout the structure. All loads in an infinitely rigid structure would be point loads and the structure would have to be so massive as to be useless. The pyramids of Egypt don't flex much, but they can't fly. Sit in a window seat on your next commercial flight and watch the wings flex in response to gust loads—a beautiful example of a flexible structure with rigid joints. It is the joint assembly that must be rigid and not the structure, of which the joint is only one part.

SPS TECHNOLOGIES

SHEET 1 OF 2

DRAWING NUMBER SPS-B-71120

REVISION (1) 12-13-71



BASIC PART NUMBER	THREAD SIZE	A DIA	B DIA ±.005	C MIN	D DIA	E MIN	H	J DIA	M	N DIA	Q ±.005	R MAX	R MIN	U REF	W NOM	CONCENTRICITY Y	CONCENTRICITY Z
BE 1218-3	.190-32	.375	.166	.278	.250	.130	.260	.100	.090	.047	.045	.025	.015	.04	250	.011	.006
BE 1218-4	.250-28	.438	.223	.348	.312	.168	.330	.150	.090	.070	.065	.025	.015	.04	.312	.013	.006
BE 1218-5	.312-24	.500	.281	.419	.375	.176	.375	.206	.090	.070	.075	.025	.015	.05	.375	.015	.006
BE 1218-6	.375-24	.562	.344	.490	.438	.184	.422	.260	.090	.070	.095	.025	.015	.05	.438	.017	.006
BE 1218-7	.437-20	.688	.401	.563	.500	.212	.490	.310	.090	.070	.115	.030	.020	.06	.500	.019	.006
BE 1218-8	.500-20	.750	.463	.633	.562	.250	.530	.370	.090	.070	.130	.030	.020	.06	562	.023	.006

NOTES:

1. MATERIAL – INCO 718 PER AMS 5663
2. HEAT TREATMENT – 180,000 PSI MINIMUM, TENSILE STRENGTH AT ROOM TEMPERATURE
3. PLATING – NONE
4. FLUORESCENT PENETRANT INSPECT 100% PER MIL-I-6866, ACCEPTANCE CRITERIA PER SPS-1-650
5. CONCENTRICITY
   HEAD O.D. TO SHANK DIAMETER WITHIN "Y" T.I.R.
   THREAD P.D. TO SHANK WITHIN "Z" T.I.R.
6. REQUIREMENTS –
   HEADS OF BOLTS MUST BE FORGED
   LIGHTENING HOLE MAY BE FORGED OR DRILLED
   THREADS TO BE ROLLED
7. * PART NUMBERS – THE PART NUMBER CONSISTS OF THE BASIC PART NUMBER, DASH NUMBERS AND LETTERS.
   THE BASIC PART NUMBER IS BE-1218.
   THE FIRST DASH NUMBER DESIGNATES THE DIAMETER.
   THE SECOND DASH NUMBER DESIGNATES THE LENGTH AND GRIP LENGTH.
   ADD "H" TO THE FIRST DASH NUMBER TO DESIGNATE CROSS DRILLED HEADS.

   EXAMPLE: BE 1218-4H 8 = .250-28 BOLT, 1.250 LONG, .500 GRIP, WITH CROSS DRILLED HEAD.

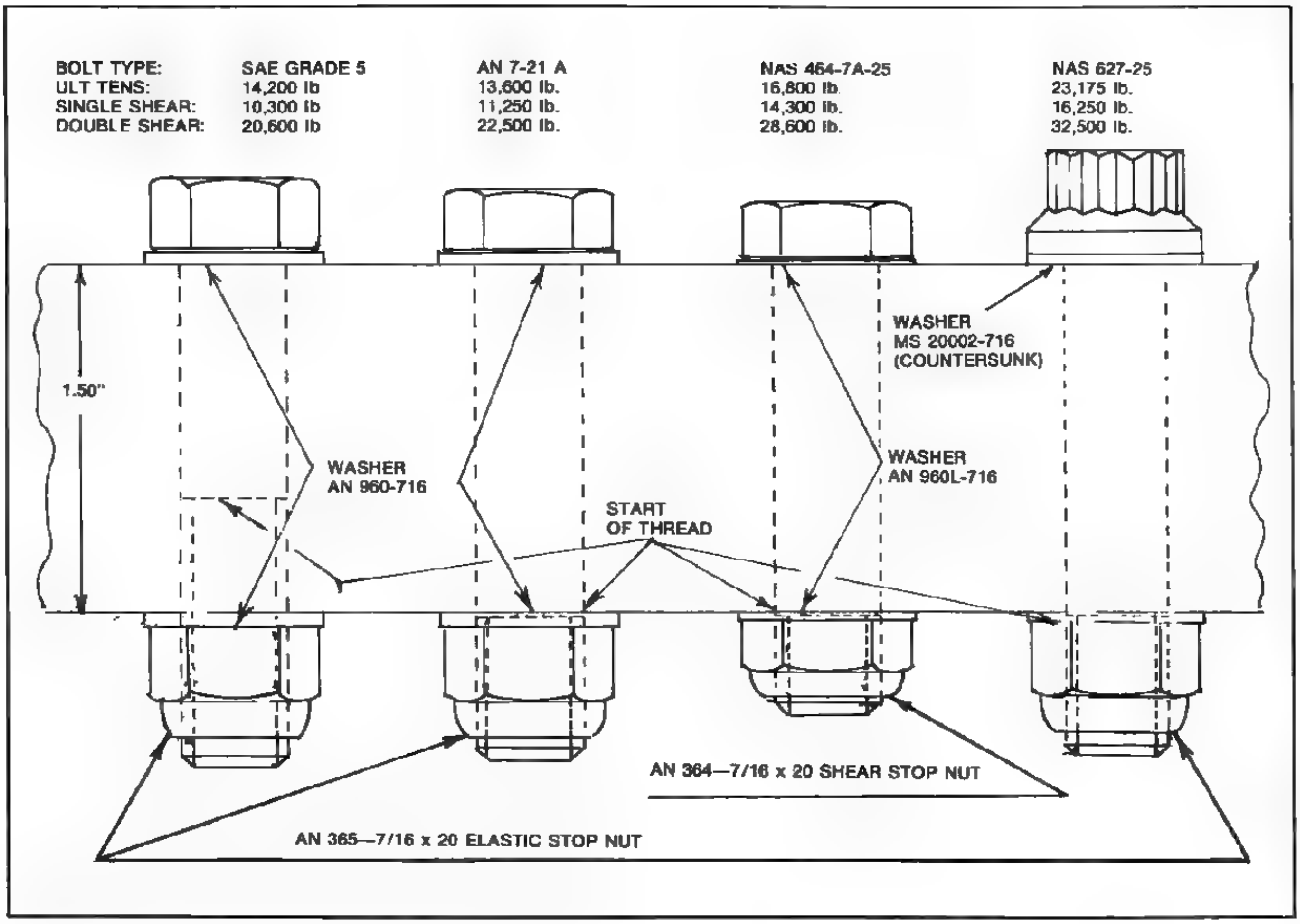
8. COUNTERSINKING OF DRILLED HOLES AT MFGR'S OPTION
9. CHAMFER "U" PLUS INCOMPLETE THREADS NOT TO EXCEED 2 PITCHES
10. RECOMMEND THIS BOLT BE USED IN CONJUNCTION WITH MIL-S-8879, UNJF-3BG INTERNAL THREAD
11. REFERENCE DIMENSIONS ARE FOR DESIGN PURPOSES ONLY AND ARE NOT AN INSPECTION REQUIREMENT
12. DIMENSIONS IN INCHES
13. BREAK SHARP EDGES .003-.015

TOLERANCES ± .010 AND ± 2° UNLESS OTHERWISE NOTED

STANDARD

STANDARDS AND SPECIFICATIONS	TITLE	
* SPS-B-640 APPENDIX 3.5	BOLT, ENGINE - 12 POINT 180,000 PSI MINIMUM TENSILE STRENGTH FOR APPLICATIONS TO 1200°F. INCO 718 MATERIAL	DRAWN BY TMcG DATE 9-18-68
FED. IDENT. CODE NO. 56878		APPROVED [signature] DATE 12-17-68
CUSTODIAN: JENKINTOWN, PA.		PART NUMBER BE 1218

BE-1218 engine bolt specifications.



Nut and bolt combinations for 1½ in. work thickness.

5

Bolt specifics

Hex-head bolt versus socket-head cap screw

What we generally call an Allen bolt is correctly termed a socket-head cap screw. It is called an Allen bolt for the same reason that stainless-steel braid-protected flexible hose is called Aeroquip and that refrigerators in my youth were called Frigidaires. The Allen Manufacturing Company was the first mass producer of quality socket-head cap screws. It is still among the few that I trust.

The socket-headed cap screw has two advantages over the standard hexagon-headed bolt: First, the head takes up less space, and second, high-quality socket-head cap screws are more readily available than are high-quality hex bolts.

But there is no free lunch and, for many of us, particularly for the racer, they also have three disadvantages: First, they have very small heads. If they are to be loaded in tension, the limited bearing area under the head will prevent tightening the bolt sufficiently to take full advantage of the strength of the alloy—unless, of course, a hardened washer is inserted under the bolt head. Second, they are manufactured with very long threads. If they are to be used in shear (loaded in bending), the excess thread must be removed before they can be properly installed. And third, the heads are case hardened and are therefore difficult to drill for safety wire.

For shear applications, where you almost certainly will not tighten the bolt to the level of stress that would take full advantage of the strength of the bolt material, there is no performance difference between using a socket-head or a hex-head. On racing cars, I use hex-headed AN bolts whenever possible for reasons that I have stated (many times). If I don't have access to AN bolts, I usually use modified socket-head cap screws because they are likely to be both easier to find and superior to any of the available alternatives. But, the lack of bearing area under the socket head can lead to big trouble when the bolt is stressed in tension—if we do not recognize the potential problem. God arranged things so the unwary and the ignorant will get caught out. We call it Darwinism.

We can readily obtain high-quality socket-head cap screws because the machine tool industry uses them to hold machine tools together. They do so because the heads fit into relatively small counterbored holes, and thus allow parts to slide past each other without snagging on the bolt heads. The machine tool industry also uses hardened and ground machine washers under the heads of their socket-head cap screws. If they did not, they would not be able to tighten the things sufficiently to develop the required clamping force, let alone lock the threads. What this means is, when you decide to replace your stock hex-headed head bolts with an SPS Unbrako socket-head cap screw for a cylinder head bolt, you had damned well better use a hardened and ground washer under the head or the bolt head will sink into whatever washers you do use. The next thing that will happen is that the residual stress within the bolt will relax. The clamping force will relax with it and the cylinder head will lift. There is yet another possible scenario here. After the residual tension relaxes the bolt can loosen and break, and *then* the head will lift.

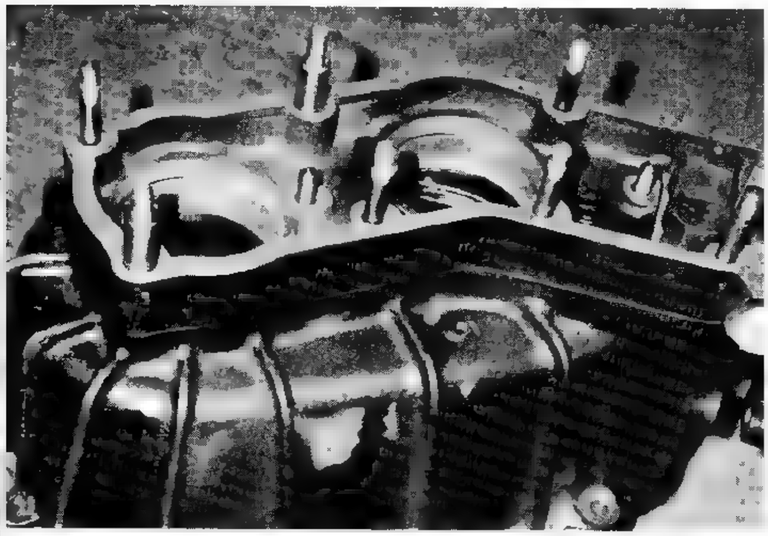
Many people consider the AN washer to be a hardened washer simply because it is an aircraft certified part. The AN washer is in no way suitable for use under the head of an internal wrenching bolt. As a matter of course, I use hardened and ground washers under the head of *every* cylinder head bolt, hex- or socket-head. Even those on kart engines. They are available from any machinery supply house, some industrial hardware stores and from your local farm equipment dealer. Yes, I mean John Deere, Caterpillar, International Harvester and others. The people who design farm equipment know a great deal about fasteners, fatigue and metallurgy and, from the metal fatigue point of view, put out some of the best designed equipment in the world. You can absolutely trust any OEM fastener that you buy from a farm equipment dealer.

Studs

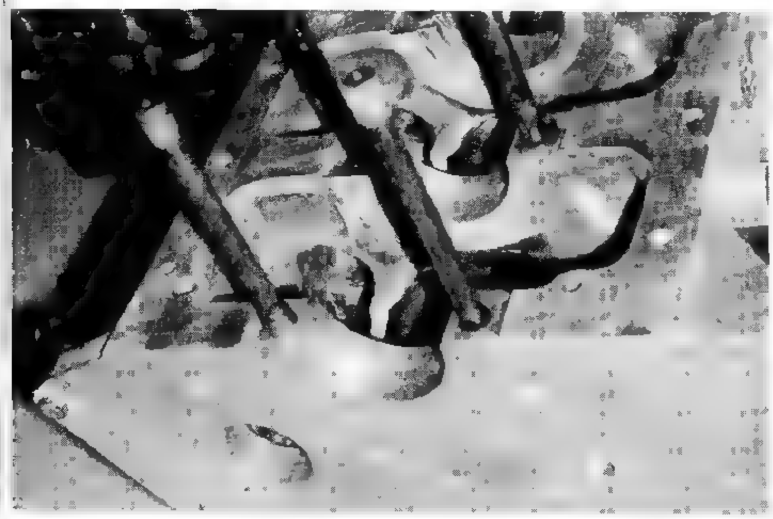
We racers use a lot of studs, particularly in the engine and in the transaxle, differential housing and bellhousing groups. Part of our rationale for this practice is that we don't want to hurt the female threads in castings by repeated removal and installation of bolts. From a theoretical viewpoint, this line of thinking is supposedly incorrect; repeated insertion is not supposed to damage proper threads. From the practical point of view, however,

we know better. But since a stud is properly inserted finger-tight, we can be certain that the stud will not damage the female threads.

I have a further reason for using studs. I can lock a stud into the casting with Loctite and then use an elastic stop nut on it. If I don't lock the stud in place and I do use a lock nut, eventually the stud will turn in the casting rather than the lock nut turning on the stud. Murphy tells me that this will happen after my driver has talked me into changing gears in the ten minutes between the Sunday morning warm-up and the race. If I use bolts, I am going to use up a mile of safety wire and hours of time. Speaking theoretically, Loctite is not supposed to be necessary and it is supposed to go away at relatively low temperatures (300 degrees Fahrenheit). But again, from practical experience, we know better. I was delighted to read, in the February 1987 issue of *Circle Track* that no less an authority than Smokey Yunik agrees with me in this respect, and studs all of his engine blocks with Loctite.



Tubular location dowel installed concentric to stud.



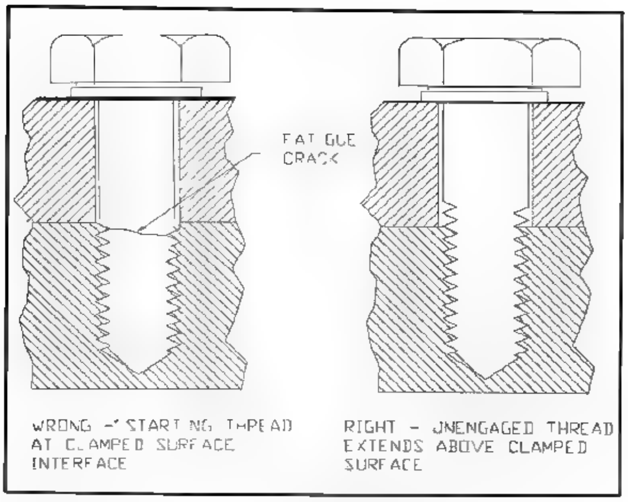
Waisted stud used to locate main bearing cap on racing engine.

With the highly stressed engine fasteners—cylinder head, main bearing caps and connecting rods—studs are actually structurally preferable to bolts. There are two reasons for this. First, since the stud is threaded finger-tight into the female threads and the nut is tightened onto the stud, less of the force of tightening is used up in overcoming thread friction, and so the desired level of residual stress can be more accurately approximated. Second, for the same reason, residual stress in a stud is more evenly distributed than that in a bolt, resulting in a more efficient installed fastener. Yes, this does mean that the common practice of torquing a bottomed stud into its hole is wrong—very wrong. Studs should never be forced into the bottom of their holes; they are meant to be installed finger-tight. Again, in low-stress applications, we often break this rule for reasons of practicality.

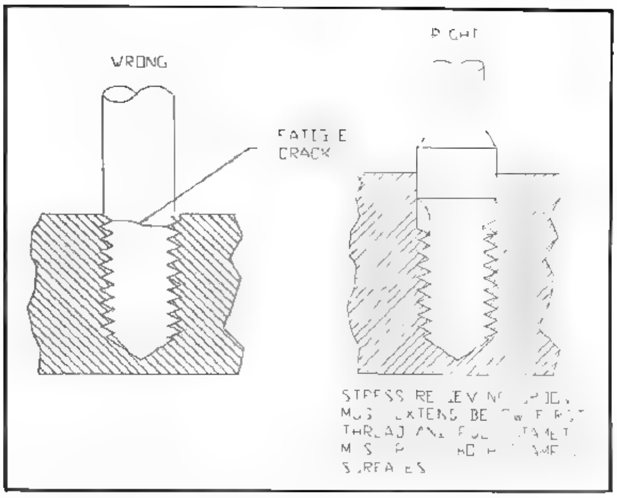
Racers do, however, have an unfortunate tendency to use studs as locating devices and this we should not do. Studs, like bolts, are meant to be clamping devices only. Location should be by means of dowels or by locating or piloting diameters on the clamped parts. A locating dowel can be a hollow cylinder surrounding a stud just as easily as it can be a solid one in a separate hole.

Me, I like studs. As fasteners, studs make assembly easy because, even if the part to be installed cannot be located by the studs, it can be guided into place on them. This is of particular interest when the part is hot and heavy, like a Ford 9 in. differential. It is of equal interest when the part is awkward, like a Hewland transaxle, complete with rear suspension assembly, which we are trying to stab into the clutch spline and onto the bellhousing. In many cases, though, we do the stud thing all wrong.

First of all, a stud properly requires a slight interference fit between the male and female threads; otherwise, it will eventually back off while the nut is being loosened. This used to mean that we, at least theoretically, had to use a Class 5B tap to form the female threads in a hole that was going to receive a stud. Nobody ever did this, including me. I have never seen a Class 5 tap. I use Loctite to achieve my interference fits. I should point out here and now that if a cylinder head stud is going through into the water jacket I use Loctite pipe sealant with Teflon paste and nothing else. Unlike the other sealants, the Loctite paste is not only an excellent sealant, it also offers some thread-locking properties. I have never been accused of being a trusting soul, so I also use some form of threaded insert for any female thread that is going to see much disassembly. More on both inserts and Loctite later.



Correct and incorrect locations of threads in relation to joint face.



Correct and incorrect stud thread location and hole design.

When there is going to be little stress involved and you are certain that the stud will do no locating, then the standard practice of tightening the stud into a tapped hole so that the run-out thread of the stud ends up at or slightly above the surface of the hole borders on being acceptable. The problem is that tightening the nut on the stud is bound to pull up the material at the edge of the hole. This material will form a ridge around the stud which, in turn, will prevent proper clamping of the mating surface. Worse yet, if the stud is going to do any locating at all, the locating will put a bending load on the stud—at the joint interface. If you locate the run-out thread at the joint surface, then the shear stress from the bending load, when added to the residual tension stress already present in the tightened stud, ensures you that you will eventually achieve a fatigue failure—right at the run-out thread.

The solution to this problem is to counterbore the hole so that tensioning the stud cannot raise a ring of parent metal around the hole, and so that the run-out thread is removed from the danger area. Since the root of the run-out thread is liable to be the most severe stress raiser in any threaded fastener, you should relieve the run-out thread. The stress-relieving groove at the last thread must extend below the first female thread when the stud is bottomed.

Selection of studs

Most of the aftermarket hot rod supply houses now offer high-strength engine stud kits. Be careful here! If the threads are lathe cut, don't even consider the product. Likewise, if the threads are not to UNJ or UNR specs, don't use the product. If the manufacturer (or rep) brags about 4130 chrome-moly steel, cross them off your list. Threads must be straight and concentric with the shank, and minimum hardness should be 38/42 on the Rockwell C scale. There are a lot of good products in the high-performance aftermarket. Unfortunately there is also some well-advertised and merchandised moonglow. Needless to say, the same is true of the aftermarket engine bolts. There are now alternatives, however.

ARP

After years of aftermarket engine fastener hype and dubious products, someone is finally doing it right. ARP (Automotive Racing Products) of California (see appendices for address) manufactures and markets a comprehensive and righteous line of threaded fasteners for high-performance engines. These folks know what they are doing, and they are doing it without compromise. Their designs are right, their metallurgy is right (they use 8740 nickel, chrome, molybdenum steel), their heat treat is right and their manufacturing processes are right. These are the only automotive aftermarket fasteners that I can wholeheartedly recommend. They also offer a comprehensive free catalog/technical manual which every one of us should read.

While I am on that subject, many people feel that a twelve-point head is proof positive of a high-quality bolt. It is not, and neither is a socket-head or a reduced hex-head. Some of the hypsters have taken advantage of this belief and are offering twelve-point junk bolts (as well as internal wrenching bolts). There is absolutely no magic involved in bolt head design or manufacture. Anybody with either a hot or cold heading machine can forge any shape head they want to onto any blank. The twelve-point head was developed in aerospace because they needed a bolt head that would accept a small socket wrench and not strip at the installation torque required by high-strength bolts.

A practical view of the bolt picture

Everything that I said so many years ago in *Prepare to Win* about bolts and their relationship to the racing car stands. Judging from your letters, I seriously underestimated the difficulty of obtaining aircraft hardware in the hinterlands. Alternatively, I may have overestimated the ingenuity and determination of the racer—but we won't go into that. Living and working in southern California does have its compensations. I still strongly prefer the AN, MS and NAS items, and in many instances I insist on them. But I do spend a lot of time in East Nowhere and I do recognize reality if my nose is rubbed in it often enough.

If you cannot find the right stuff locally, here are a few acceptable alternatives that I probably should have listed originally.

Earl's Performance Products, The Dillsburg Aeroplane Works, and Aircraft Spruce and Speciality maintain stocks of AN/MS/NAS hardware and offer UPS service. They each offer excellent catalogs. Earl's and Dillsburg are a lot easier to deal with on the telephone.

Since most racing cars are designed around junk bolts, any premium grade internal wrenching bolt will do the job in just about any application—with the exception of highly stressed engine bolts—like cylinder head, connecting rods and flywheels. In my book, "premium grade" must specify UNR threads and include: Standard Pressed Steel's Unbrako line, the best industrial bolts that I know of. There are several members of this family. One thing to be a little careful of, though, is that the K16 type, while it is rated at only 150,000 psi UTS, has twice the *fatigue life* of the standard Hi-Life Unbrako which is rated at 190,000 psi UTS. Remember what I said, hard sometimes means brittle. My book also includes Allen Manufacturing Company's 1960 series socket-head cap screws and Holo-Krome's thermo-forged line of socket-head cap screws.

As a generalized tip, the manufacturers of *good* Allen bolts always describe their products as socket-head or internal wrenching cap screws, and the good bolts use the UNR thread form, which you can detect with the naked eye. If you cannot find one of these brands where you live, then I can only suggest that you move! The best sources are machinery supply houses and industrial hardware stores.

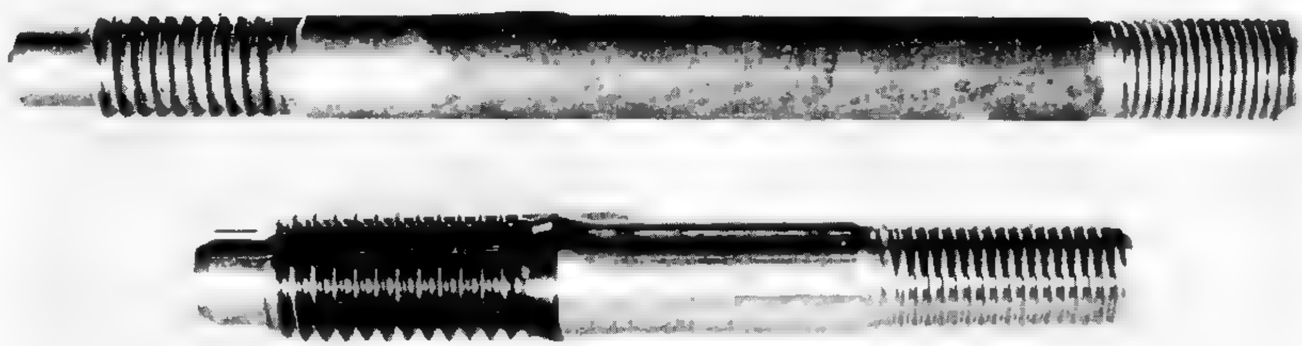
The basic problem here is the inordinate amount of work required to convert even a good industrial tension bolt into an acceptable race car double shear bolt. The process is as follows: first, find a bolt with the required diameter, thread pitch and an acceptable grip (unthreaded shank) length.

Second, cut off the surplus thread. Leave sufficient thread to fully engage the threads of the nut plus a minimum of three full threads. Standard AN thread lengths were listed in an earlier chart.

The surfaces of socket-head cap screws are harder than the hinges of hell and will wreck even the best hacksaw blade in short order. I use a pneumatic die grinder with a thin metal removal wheel.

Third, chamfer the cut-off end so that you can screw a nut onto it without cutting the nylon locking ring. I use, in descending order of preference, a belt sander, a disc grinder, a grindstone or a file. I am too lazy to chuck each bolt into a lathe.

You may also get to drill the thing for safety wire. Since all internal wrenching bolts are case hardened, they are every bit as destructive toward drill bits as they are toward hacksaw blades. The tip



The wrong stuff and the right stuff: hot-rod aftermarket head stud, top, and the top-quality ARP head stud, below.

here is to grind a small flat on each side of the bolt head in the area of the hole-to-be and to seriously center punch a starting pocket. It helps the temper of both the operator and the drill bit to grind the flats approximately parallel to the corresponding flats of the internal hex, to use a sharp cobalt drill bit, the right drill speed and a lubricant. It also helps (a whole lot) to use Tuck Jones' drill jig.

Everything that I have just said about socket-head cap screws holds true for SAE Grade 8 hex-headed bolts except that first, they are typically not as strong as first-quality socket-head cap screws (150,000 psi versus 180,000 psi); second, quality control in manufacture tends to be spotty; third, they are not hard enough to hurt either hacksaw blades or drill bits; fourth, they are not available with UNR threads; and fifth, they are often counterfeited.

Noncritical bolts

A great many of the bolts and machine screws on the typical racing car fall into the noncritical class. You are not going to break the #10–32s that hold the mirrors on, the 1/4–28s in the fuel cell access panel or the 5/16–24s that serve as throttle and clutch stops. The 7/16 in. bore rod end bearings so loved by many small formula car constructors don't require NAS bolts for the simple reason that the loads involved are too low to cause a problem if you were to use a 7/16 in. nail. This means that there are many applications where, when nothing better is available, an SAE Grade 5 (three line) bolt will do the job—but not on one of my cars!

For a realistic indication of what is actually required—assuming top-quality hardware, properly scheduled and performed maintenance and scheduled replacement of parts—take a good look at the bolts on a current Brabham, Williams, McLaren or Beneton Formula One car—assuming that our provincial sanctioning bodies/promoters haven't killed Formula One in this country. For an equally valid indication of what is required to do the job on a good amateur car, take a look at the latest Swift brainchild of David Bruns. For reasons involving quality control (or, rather, lack of it) and price, I don't much like using SAE Grade 8 bolts but I will use them in a pinch. In insist on seeing the box that they came in, and I like to see the invoice from the manufacturer. I still refuse to use or keep in stock super-whatever and superior to SAE Grade 8 bolts.

Common sense in bolt selection

We should be sensible about this business of bolts and the racing car. If your race car arrived from the manufacturer with an aircraft bolt (or a good socket-head cap screw) holding a particular widget on, you can bet that it is there for a reason—the builder does not enjoy spending money. You will replace it with an inferior bolt at your peril. On the other hand, replacing a British Allen bolt with an SPS Unbrako socket-head cap screw, or replacing an original grade nothing bolt with a modified SAE Grade 8 or an AN bolt is not only perfectly safe, it is a step in the right direction. You have to use your head!

My standard practice is to keep only MS bolts in open stock (that is, the stock that anyone other than myself can get at). That way I know that no one is going to substitute an inferior bolt without my knowledge. I know there is no realistic way that I can foresee every fastener need that is going to come up in the course of a season, even if I had room to carry them. So I also carry, in my own private stock, a pretty comprehensive selection of SPS socket-head cap screws and SAE Grade 8 bolts in various diameters and grip lengths. I then modify to suit when it becomes necessary.

I also keep in stock a few ridiculously long bolts in each diameter so that, when the need arises, I can break my oft-quoted cardinal rule about never die cutting threads onto a bolt. I only do it when I absolutely have to, and then only with bolts loaded in double shear. And I do replace the thing at the first opportunity. One last word on the subject: I would much rather use a modified Grade 8 bolt than die additional threads onto an NAS bolt!

Stainless, titanium and aluminum bolts

The aerospace industry produces bolts in stainless steel, titanium and aluminum alloys for specialized applications. I use the aluminum units to save weight in noncritical situations—inspection panels, fuel cell closures and the like. Contrary to popular belief, the stainless items are useless for our purposes simply because they are weak (about 90,000 psi UTS) and should be used only on boats, and then with a great deal of caution.

I don't use titanium fasteners, partially because I can't get them, partially because they are notch sensitive and partially because titanium tends to gall on steel. Among the things that I do not need is a bolt stuck in its hole or onto its nut.

Dutchman tubular clamp mount.

Caution: Peligro! Danger!

There are some applications where it is just not safe to use anything other than the best bolts available. These applications, regardless of what came on the car, include but are not limited to the following.

Brake disc to top hat bolts. Having once witnessed the results of the shearing of these items—and not having liked what I saw even a little bit—I use 180,000 psi NAS-624 series twelve-point bolts with proper NAS beveled washers under the bolt head and high-temperature jet or K nuts. If I cannot get the NAS bolts, I use modified socket-head cap screws. Do not allow threads to bear against the holes in either the disc or the top hat. And *do not use stainless bolts in this application!* They will stretch when exposed to brake heat. If you should be so fortunate as to be working with carbon/carbon brakes, you will be forced to use extremely high-temperature fasteners. They are available in temperature ranges up to 1600 degrees Fahrenheit. Check with the manufacturer of your brakes or with Tilton Engineering for specifics.

Crown wheel to differential housing or spool bolts. A fair percentage of the failed Hewland crown wheel and pinion sets that I am called in to autopsy have failed because the crown wheel (ring gear) loosened on the carrier. The results of a loose crown wheel are both predictable and catastrophic. Hewland assembles the things with bolts that are marginally satisfactory for one-time use, and then uses soft iron lock tabs to hold the bolts in place.

The place bolt.

This is insanity! Even though the Hewland manuals specifically warn against reuse of the bolts, many people do use them over again. Some people even get away with it—for a while. The shock loads in shear are fearsome, but the tension loads due to the action of limited-slip differentials are worse. Eventually, either the bolts stretch or the lock tabs squeeze out. This makes the ring gear just a little loose on the carrier and away we go.

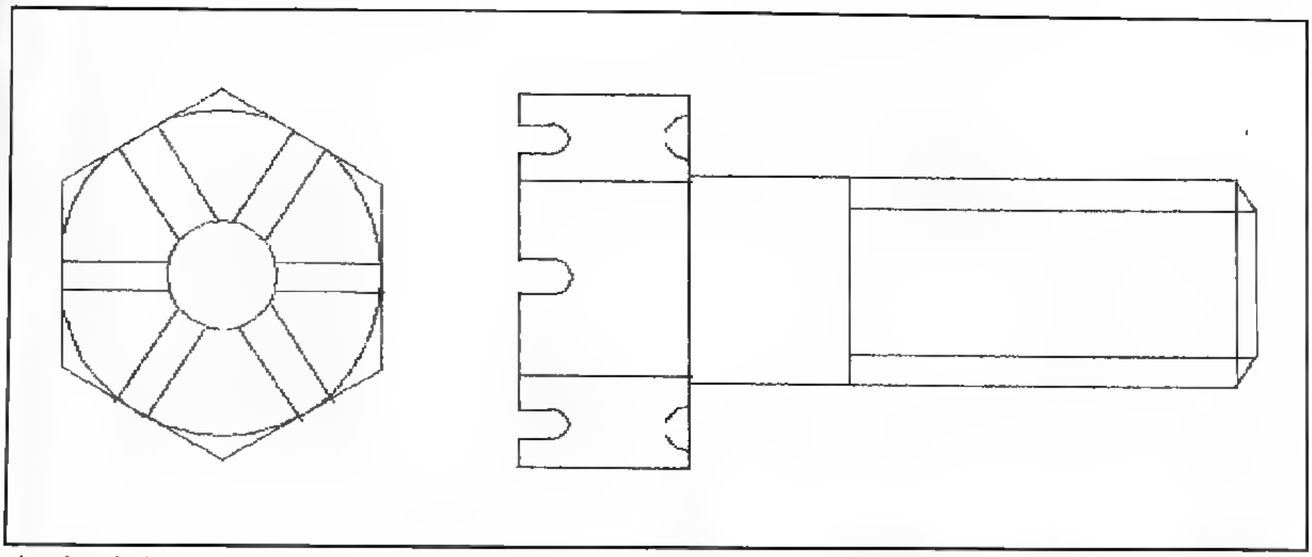
Another transaxle scenario that I have seen is where the loosened bolt fatigues from the imposed bending load and breaks off. Stress raisers being what and where they are, the bolt breaks at the root of the run-out thread. This leaves just enough shank length in the hole so that the bolt head and attached broken shank cannot escape (due to the proximity of the Hewland side cover). Instead, the head of the bolt becomes a dull but effective flycutter—until it either destroys the side cover or removes enough of same so that it can escape—and instantly wedge itself between the pinion and the ring gear. In either case, end of deal!

The solution is to use a Ford Motor Company bolt (part number B8AZ-6379A, Bolt, Fly/whl). This is Ford's standard V-8 manual flywheel to crankshaft bolt. It is a place bolt, seldom mentioned these days. As nearly as I can determine, it was developed to enhance the resistance to shock loads in the reciprocating steam engine. The head of the bolt is shaped so that, when the bolt is shock loaded in tension, the head acts somewhat like a spring in compression—a shock absorber in the classic sense of the term (as opposed to the current automotive sense which is grammatically incorrect). Automotive shock absorbers do not absorb shocks. The suspension springs do that. The shock absorbers dampen the motion of the springs, which is why the rest of the world calls them dampers.

The place bolt has a counterbored bearing surface and a slightly hollowed and/or slotted head. The bolt head is dimensioned so that it is stronger than the shank and proportioned to act as a spring. When a shock load arrives, the head compresses and absorbs the shock. Clever!

This technique is still being used in the automotive industry. If the bearing surface of a bolt head is undercut just enough so that it will elastically deform to a true bearing surface when tightened, then the residual stress in the head will add to the residual stress in the bolt shank to improve the locking capacity of the assembly. What has been created is a super-duty conical lock washer.

To be used in a Hewland, the Ford bolt must be shortened about ⅛ in., drilled and then chamfered for lock wire. It will then fit all of the Hewland ring gears and differential carriers, and it *will not fail!* I reuse these bolts as long as there is no visible thread damage in the form of flattened thread crests—I have never seen one of these bolts elongate. The bolt has a larger than normal hex head (¾ in.

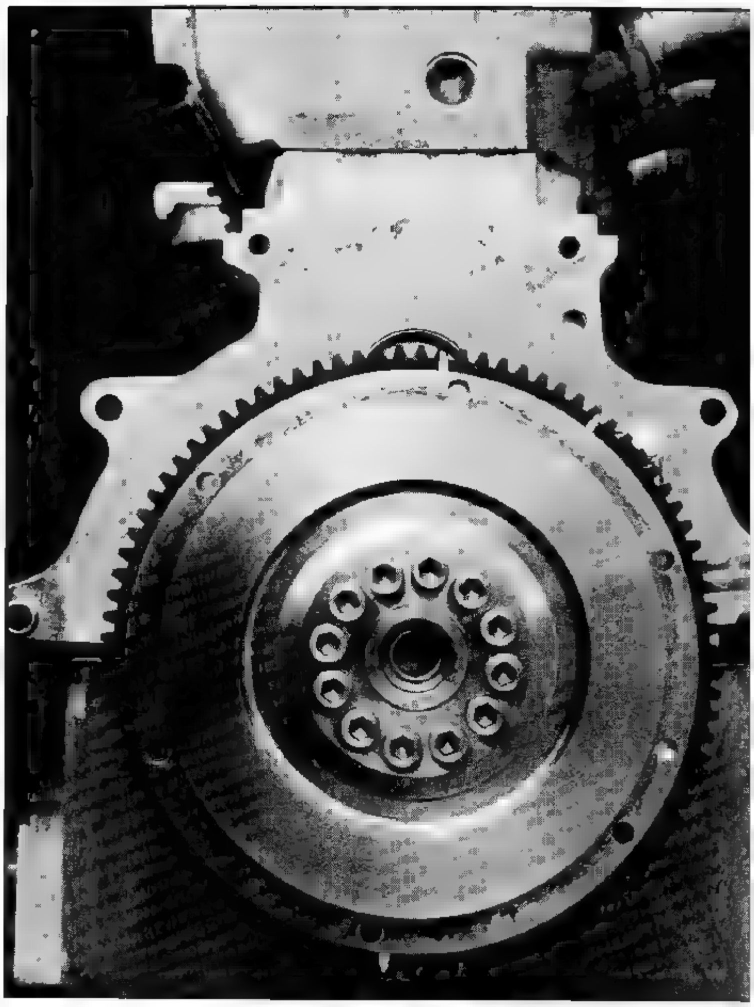


The place bolt, top and side views.

instead of ⅝ in.). Ideas do not have to be new to be good—there is nothing wrong with old technology. But, of course, it wouldn't do much good to use a place bolt with a standard soft steel washer. Since the differential carriers are plenty hard, I don't use any washers at all—just red Loctite threadlocker #261 (or, when I can find it, #272) and safety wire. The wire is not there to prevent the bolts from loosening, but to (hopefully) retain them in place if they should loosen or even break. Actually, since I have been using these bolts in this application for twenty years and have never had one loosen, the wire is there because of cowardice and force of habit.

Flywheel to crankshaft. Nothing good has ever been reported about the flywheel parting company with the crankshaft! On the V–8s I use the same Ford bolt described earlier. For the Cosworths, I use a suitably modified SPS Unbrako socket-head cap screw—with the correct grip length. This is a pain in the butt. Not only must I remove a bunch of thread and then dress the end of the bolt, but I must also shorten the head to gain clearance for the clutch. These bolts cannot be reused (not even once!), and the shortened inhex will occasionally strip when you are trying to remove them. When this happens, heat the bolt head red to kill the Loctite, hit it hard at the root of the inhex with a punch, wait for it to cool and then remove it with an Allen key or chisel (effective technique courtesy of my favorite engine builder, Steve Jennings). On the other hand, the flywheel *will not come off!* This last statement cannot be made if other bolts are used. Take your choice. ARP makes some flywheel bolts and they are excellent.

Suspension pivot bolts. Be concerned with any suspension pivot bolt with a diameter of less than 5⁄16 in. on a car weighing 900 lb. or less, ⅜ in. on 1,500

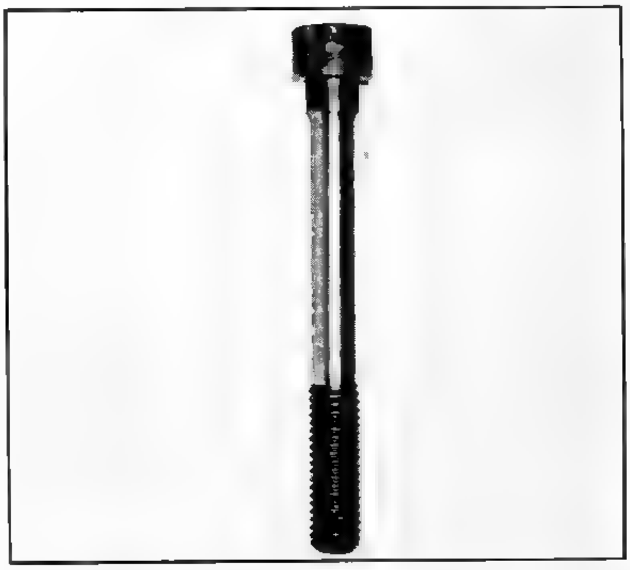


Modified SPS Unbrako socket-head cap used as flywheel bolt in Jennings/Cosworth BDA Formula Atlantic engine.

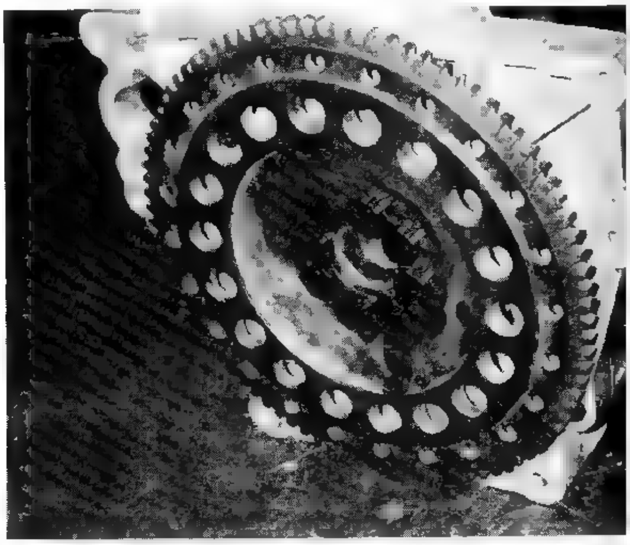
lb. cars or 7/16 in. on cars up to 2,500 lb. When cars get heavier than that, you are out of the league that I prefer to play in. My feeling is that the behemoths are liable at some time in their lives to run on the high banks and that therefore they deserve the best suspension pivot bolts that can be found.

Shift linkage universal joint to tube/rail bolts. This may sound like a strange application for first-class bolts. It is not, for the reason that no one, to my knowledge (except for David Bruns), designs cars with both a decent shift linkage layout and proper universal joints. Why this should be I do not know. It just is!

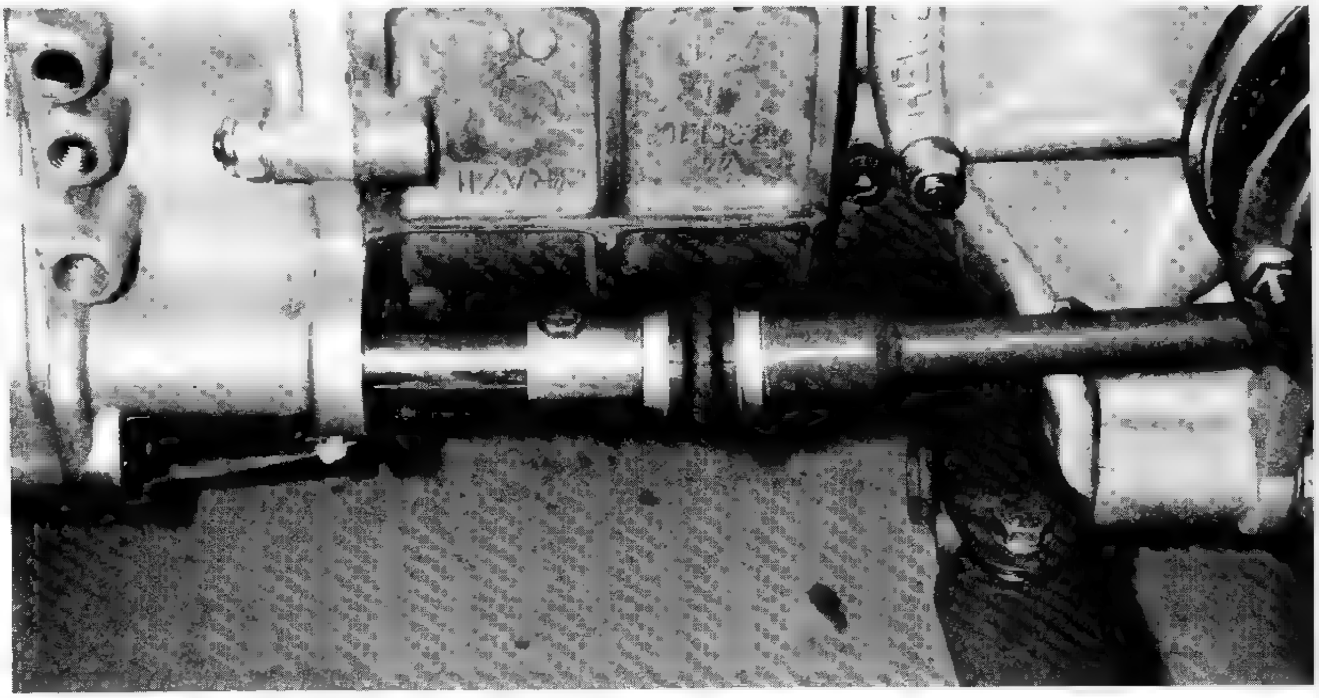
From Formula Ford to Formula One, most shifting problems can be traced directly to less than optimum shift linkage. Most racers never do anything about it. When everything loosens up and



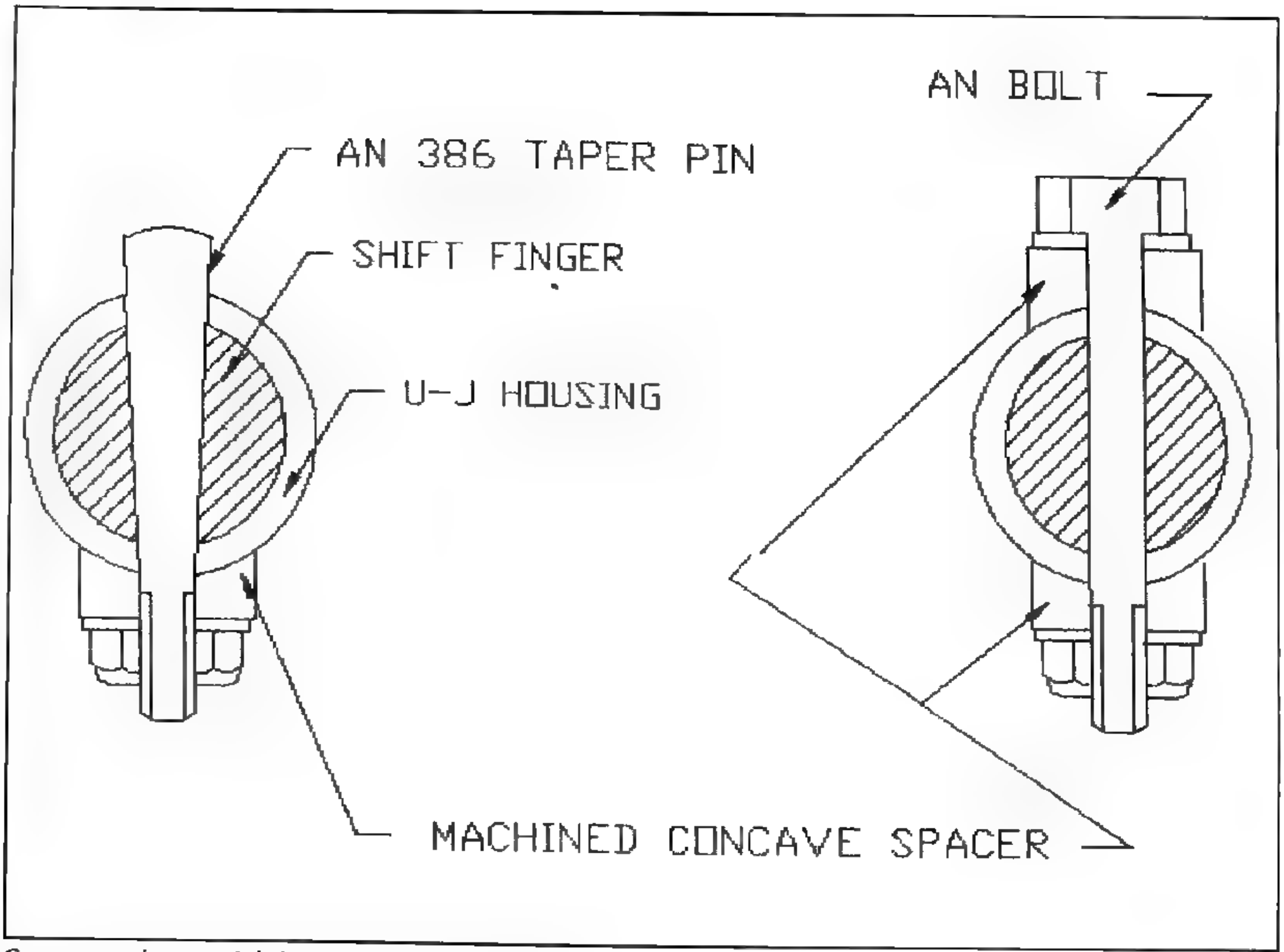
Detail of SPS Unbrako socket-head cap screw modified for use as fatigue-resistant head stud.



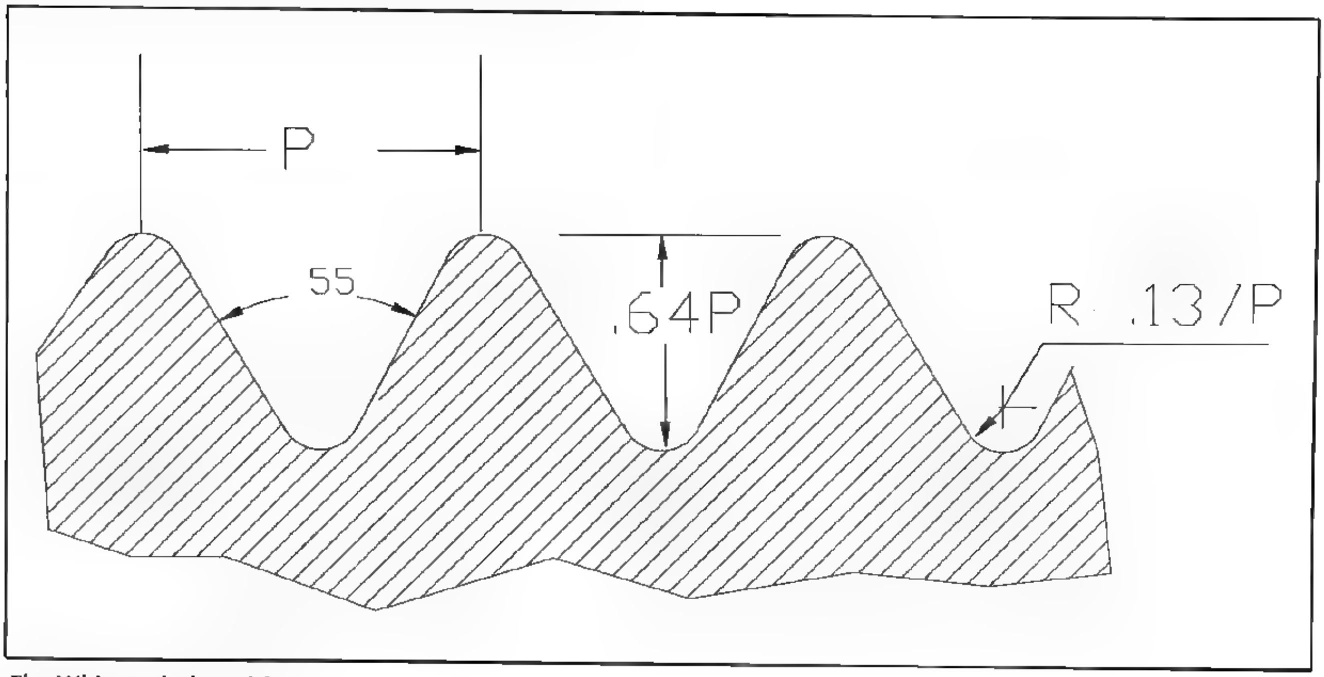
Lightened flywheel retained by ARP special purpose bolts.



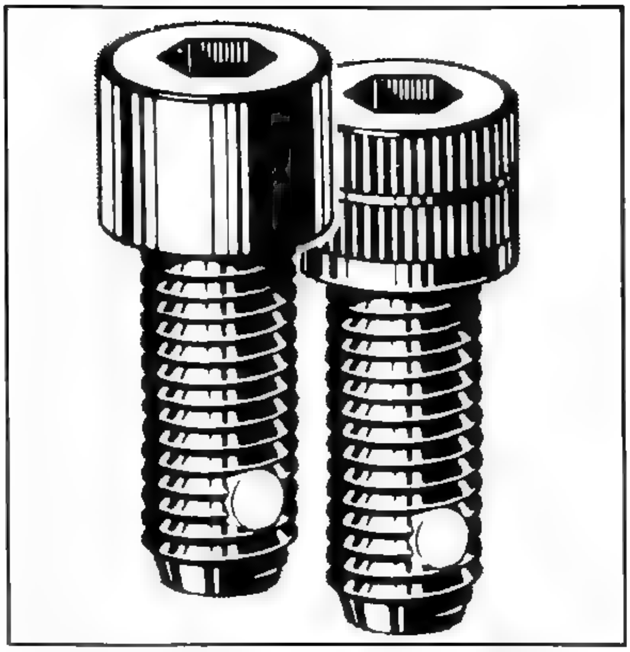
Correct attachment of shift linkage universal joint to shift rod—an excellent approach to a double shear trunnion.



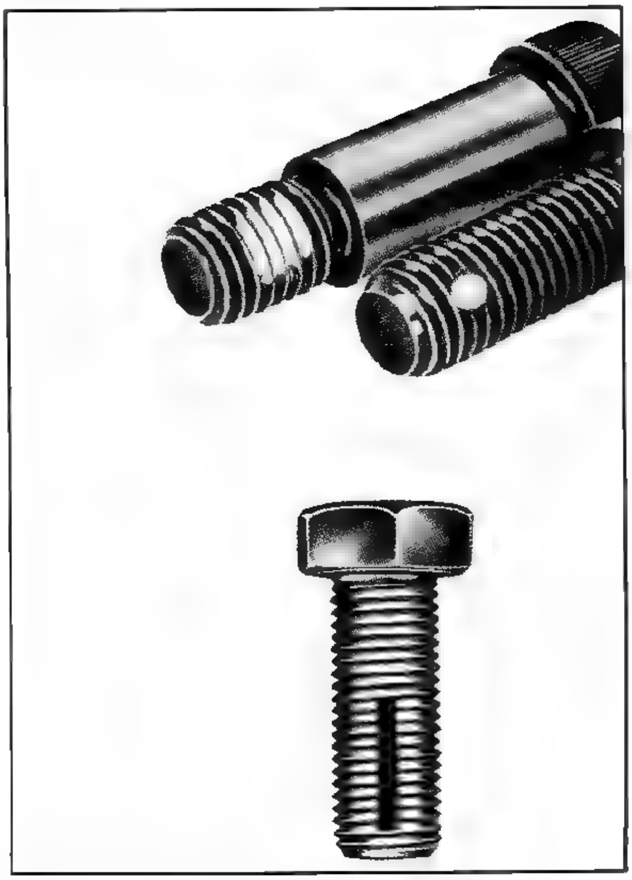
Correct attachment of shift universal to shift finger.



The Whitworth thread form.



The Allen nylock Spot-Lok socket-head cap screw.



Types of nylon-insert self-locking bolts.

becomes sloppy and horrible, the normal answer to the driver's piteous complaints is to slot the cheap and nasty universal joints that came on the car and to really graunch down on the ¼ in. through bolt that secures the U-joint to the rail. If he is feeling particularly conscientious, the operator will even drill and deburr a hole in the U-joint at the end of the hacksaw slot so that the whole thing won't split in two starting at the jagged end of the saw cut.

By now the hole is almost certain to be oversized, so the bolt is a loose radial fit. Eventually the bolt gives up and breaks. If you are lucky the bolt breaks while the nut is being tightened after a gear change and the result is a couple of skinned knuckles and a new bolt. If you are not lucky, the mechanic ignores it when the bolt "feels funny." He says nothing and does not replace the bolt. The bolt breaks on the racetrack and the result is a DNF. The use of a twelve-point NAS bolt and a shear stop nut will prevent the latter occurrence because the nut will strip before any harm can be done to the bolt. The right way to do the job is to use an Apex universal joint and an MS taper pin (described in chapter eight). A practical alternative—and the next best solution—is to use the NAS bolts with radiused washers under both the bolt head and the nut.

Trivia

Some years ago, I was in a conversation with Keith Duckworth about connecting rod bolts. Duckworth is the founding father and resident (now consulting) genius at Cosworth Engines. He made a statement to the effect that we once had the perfect screw thread form and then threw it away. He was referring to the much despised (in this country) Whitworth thread. Actually, it is not the thread form that a generation of American mechanics learned to despise (they didn't even know that it was different), but the nut and bolt hex sizes which differ from ours and simply do not make sense to us (like other British habits such as a preference for cold toast and almost raw bacon).

When Whitworth designed his thread form 150 years ago, he was thinking hard about fatigue in bolts under tension, so he designed radiused thread roots and peaks into the system. The drawings shown here indicate that he knew what he was talking about. In the praiseworthy interest of international unification (ignoring, of course, the metric system—after all, Germany and Japan had lost the war and the French don't count) we (the United States, Great Britain and Canada) shelved Whitworth for UNF and UNC in 1948.

Interestingly enough, SPS introduced their Hi-R thread form some years ago in order to reduce stress concentrations and improve fatigue life in highly stressed aerospace fasteners. A comparison of the Hi-R profile with the Whitworth thread form looks like one more example of the reinvention of the wheel.

Locking the male thread

Most of our bolt applications are loaded in shear and use a nut. In this case we always lock the female thread because it is simpler and more economical to do so—nuts are cheaper to manufacture than bolts. (We will cover self-locking nuts later.)

Every so often we run into a tension or a blind shear application where we need to lock the bolt in place at a predetermined level of internal stress. Normally we achieve this goal with Loctite and apply a safety factor with safety wire. Sometimes, however, we cannot fit safety wire and do not wish to entirely trust Loctite.

While virtually no one stocks bolts with locking threads, there are several methods of applying a mechanical lock to male threads. Several manufacturers, most notably the Allen Manufacturing Company, offer a full line of socket-head cap screws with nylon locking elements inserted along the male threads. They function exactly like the nylon elastic stop nuts described earlier. Some large industrial fastener houses keep a reasonable selection in stock. Unfortunately, my experience has been that these elements are usually ineffective as locking devices.

Several corporations will apply what is termed a pressure activated microencapsulated adhesive coating to male threads—a sort of preapplied Loctite. As near as I can determine, while this process is invaluable in production applications, it is of no interest for us since applying Loctite on small numbers of bolts is not a big deal. As a point of interest, and in case some of you are production oriented, the leader is the Nylock Fastener Corporation of Rochester, Michigan. They call their process Nylock Precote and it is available in three formulations. 3M has a similar coating.

The Long Lock Corporation of Los Angeles, insets nylon strips into bolts and screws and stocks a wide range of parts. They also do a clever thing called Dyna-Thread, in which they machine a cavity into the end of the threaded bolt shank and hydraulically expand a couple of threads to form a prevailing torque lock bolt.

Accurate Automatic Parts Inc. of New Berlin, Wisconsin, offers a line of locking studs in which the first 1½ threads are made to standard Class 3A specifications so that the stud can be easily started in a standard Class 3B tapped hole. After the first 1½ threads, the minor diameter of the stud threads is increased to provide an interference fit which effectively locks the stud in place. The degree of interference is varied to suit the tapped material. Clever—and I wish that I could buy them in my sort of quantities.

SPS offers a couple of high-tech self-locking bolts that function by the expansion during manufacture of a number of threads in the center of the threaded length of the bolt. These expanded threads work like an elastic stop nut in reverse.

While one would have to be pretty desperate to consider any of these alternatives (for reasons of both cost and convenience), it is not a bad idea to be aware of their existence. We do occasionally become desperate.

Rules of threaded fastener use

The methodology of threaded fastener use tends to be passed on by word of mouth rather than by any process of formal education. This has both its good points and its bad. The good side is that a lot of correct procedure does get passed along and it is liable to be effectively taught. The other side of the coin is that a lot of folklore gets passed along with the truth. Let's look at some of the prevalent folklore.

Myth: "An experienced and capable mechanic doesn't need a torque wrench—he can *feel* the correct tightness and repeat it as close as a torque wrench." This is obvious nonsense. The person who can feel any given torque within ten percent, let alone repeat it, has not been born. I am aware that people have been tightening bolts—sometimes critical bolts—by feel for generations. I am further aware that each of them will swear that they have never had a fastener either back out or fail in service. These people are either lucky or lying. To my mind, their success is a tribute to the overdesign of the average piece of machinery, the inherent idiot-proof nature of the threaded fastener, and the boundless ability of the human being to forget what he does not wish to remember.

The corollary to the first myth is: "Every bolt on an aircraft or racing car must be tightened with a torque wrench every time." This is an excellent idea. It is also so impractical as to be ludicrous. The truth of the matter is that every critical bolt and nut that is loaded in tension and subjected to vibration or reversals of load should be tightened to a predetermined level of stress. Determining which bolts and nuts are critical is the job of the designer and of the engineer (in the British sense) in charge. In my own case, I determine which installations require the use of a torque wrench and insist on its use.

Myth: "Bolts must always be installed with the bolt head up and facing forward so that, if the nut should fall off, gravity and the force of the airstream will tend to keep the bolt in place." There is nothing wrong with installing bolts in this manner. In fact, in the interest of standardization, I usually do. But to hope that gravity or air pressure will keep a bolt in place is unrealistic. There is a place in this world for dreamers—but that place is not in engineering. Neither is it anywhere near an airfield or a racetrack. There is absolutely nothing wrong with installing bolts wrong end up or backwards, when it is more convenient to do so.

Myth: "Elastic stop nuts are single use items. If reused, they will not lock reliably." This one started in the infancy of the elastic stop nut when the

locking collar was made from organic fiber rather than nylon and it wasn't particularly reuseable. Those days have been gone for a long time. Any of the current families of self-locking nuts can be reused many times.

In most applications the rule of thumb is: "If there is no visible damage to the thread and you cannot spin the nut with your fingers, it is OK to reuse it." Obviously this is not true in the case of critical tension applications which require high tensile nuts. I do not reuse critical nuts, locking or not. In my world this includes all connecting rod and most cylinder head and main bearing cap nuts.

Myth: "You cannot use an elastic stop nut on a bolt with a drilled shank." Why not? If passing over a deburred and chamfered hole in the bolt shank is going to destroy the self-locking feature of a nut, then I, for one, don't want to know about that type of nut.

Myth: "Always turn the nut, never the bolt." Again, it is easy to see where this one comes from, and again, it is good practice. We have seen that bolts loaded in shear should be installed in close tolerance reamed holes. When we turn the bolt in a closely fitting hole, we will produce friction between the bolt and the hole surfaces which can give a false tightening torque reading; damage the bolt surface, the hole surface or both; enlarge the hole; remove some of the plating from the bolt; and if the bolt happens to be made of titanium, the bolt will gall and weld itself to the wall of the hole.

The trouble is that there are many applications where it is easier to turn the bolt than it is to turn the nut. Some books state that, since we don't worry about turning tension bolts installed in blind holes, we don't need to worry about turning any bolt when it is more convenient to do so. Wrong! What is being overlooked is that tension bolts are properly installed in loose drilled holes where there will be little if any contact between the bore of the hole and the shank of the bolt. My rule is that convenience takes a back seat to damaging parts; I turn the nut whenever it is possible to do so. However, a primary rule of life is that we do what we have to do—and that includes turning bolts and holding nuts when the need arises.

Summary

This will be the shortest summary on record. We have discussed the what, why, how and wherefore of the threaded fastener at considerable length and in considerable detail. We have established a series of rules of conduct for their intelligent selection and use. These are rules to live by, and you will break them at your peril. But we live in an imperfect world and necessity will force you to at least bend the rules from time to time. It is my hope that understanding the reasons for the rules will allow you to bend them only selectively, intelligently and safely.

6

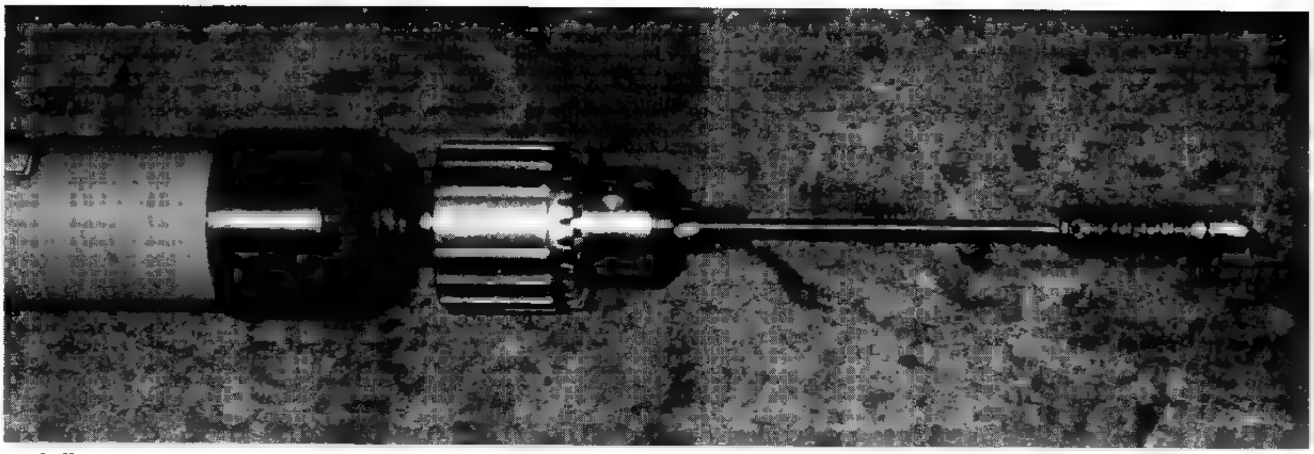
Female threads

I have spent a lot of time writing—and you have just spent a lot of time reading—about the male side of the threaded fastener picture. Most people who deal with highly stressed machinery understand and appreciate the importance of strength, toughness and quality control in bolts and studs. Possibly because they are less visible, it is more difficult to convince people of the importance of the matching female threaded parts.

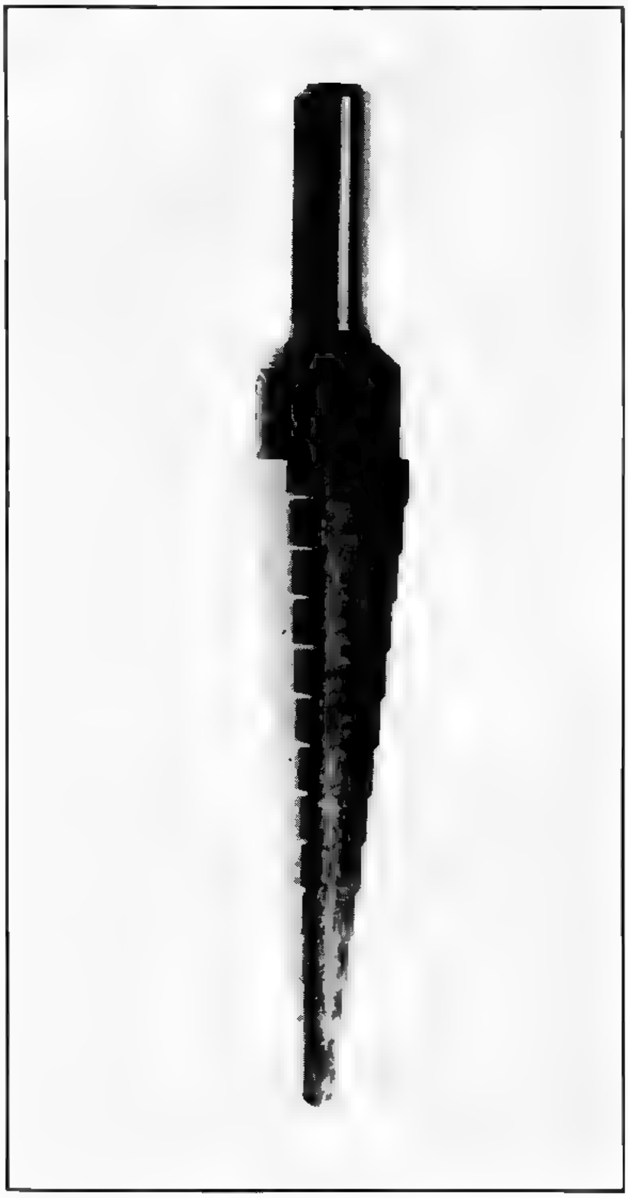
There is a slight basis for chauvinism here. It is true that, in many cases, especially when the bolt is loaded in shear, the strength of the female threaded component *is* of secondary importance. But the

THREAD	BODY DIAM	BODY DRILL SIZE/DIAM	TAP MINOR DIAM	TAP DRILL SIZE/DIAM
UNF SERIES				
#8-36	0.164	#19/.1660	0.136	#29/.1360
#10-32	0.190	#11/ .1910	0.159	#21/.1590
1/4-28	0.250	1/4orE/.250	0.213	#3/ .2130
5/16-24	0.3125	O/.3160	0.2703	I/.2720
3/8-24	0.375	V/.3770	0.332	Q/.3320
7/16-20	0.4375	7/16/.4375	0.386	25/64 .3906
1/2-20	0.500	1/2-.500	0.449	29/64-.4531
9/16-18	0.5625	9/16 .5625	0.506	33/64-.5156
5/8-20	0.625	5/8 .6250	0.568	37/64-.578
UNC SERIES				
#8-32	0.164	#19/1660	0.1324	#29/.1360
#10-24	0.190	#11/.1910	0.1476	#25/.1495
1/4-20	0.250	1/4orE/.250	0.1990	#7/.2010
5/16-18	0.3125	O/.3160	0.2559	F/.2570
3/8-16	0.375	V/.3770	0.3110	5/16 .3125
7/16-14	0.4375	7/16-.4375	0.3642	U/.3680
1/2-13	0.500	1/2 .500	0.4219	27/64-.4219
9/16-12	0.5625	9/16 .5625	0.4776	31/64-.4844
5/8-11	0.625	5/8-.625	0.5315	17/32-.5313

Tap drill hole diameter chart for 75 percent depth of female thread.



A drill stop.



The Unibit.

bottom line remains that, in engineering as in life, the male part is useless without female threads.

There are two basic types of female threaded components: the hole threaded directly into a part and the removable nut. I will begin the discussion of female threads with the threaded hole.

Tapped holes

There is actually little to say about the hole threaded directly into a part, except how to do it. Female threads are cut in a predrilled hole with a tool called a tap. We racers typically do a pretty miserable job of tapping holes. Several criteria must be met in order to end up with acceptable threads in a tapped hole: first, the tap must be sharp and undamaged; second, the hole must be drilled to the correct diameter; third, the hole must be drilled straight and in the right direction (almost always normal to the surface of the part); fourth, the correct lubricant must be used; fifth, the tap must be started straight in the hole; and sixth, the chips must be cleared frequently.

The average racer tends to ignore or at least improvise most of the above and gnaws his way through the hole like a beaver cutting a tree—and the threads show it. Two things save us from ourselves, most of the time: First, it is unlikely that you will ever tap threads in a part that will be critical; and second, the length of engaged thread is liable to be long enough to make the unit stress on the female threads relatively low.

This is hardly reason enough to cut a bad thread on purpose, however. There are a few simple rules that, if followed, will result in an acceptable thread every time.

Don't guess at the size hole to drill. If the hole is too small, you will probably break the tap. If it is too large, you will end up with insufficient depth of thread. The tap drill size chart that appears here is calculated to produce a seventy-five percent depth of thread. The increase in strength between female threads with seventy-five percent depth of thread

and those with 100 percent is approximately five percent. The difference in the force required to turn the tap is about 300 percent, and the chances of breaking the tap increase by a factor of at least ten.

Don't try to drill the hole with a hand-held drill motor. Use a drill press and make sure that the table and vise are perpendicular to the surface of your work. Better yet, use a mill if you can get at one. If the hole is to be along the axis of a cylindrical or hexagonal part, drill the hole in a lathe.

Taps are made in three styles: taper, plug and bottoming. The idea is to start the thread with a taper tap because it is self-aligning. Normal holes are finished with plug taps and the thread does not extend to the bottom of the hole. If the thread will extend to the bottom of the hole, finish with a bottoming tap. If you are going to tap a lot of holes, you should have all three styles of taps. The sets that you buy from the tool store are usually plug taps and can be started straight with care—extreme care. The taps that you can buy at the Tool Shack type places are junk and will not cut acceptable threads. The cheapest set of taps that should be considered is available at Sears; I do very little tapping so I carry a Sears set. I also carry a full set of good #10–32 and ⅜–24 taps from the machinery supply house as well as a complete set of left-handed taps. To tell the truth, I carry several #10–32 taps because I have been known to break them!

Start the tap straight. If you can, start it in a lathe, a mill or a drill press. Once they are started, taps are self-aligning. If the tap starts into the hole straight, and if the diameter of the hole is correct, it will continue straight and will cut straight threads. If it starts crooked, the best that you can hope for is that it will pull itself straight after ruining the top few threads. The worst is that you will break the tap. The in-between result is a completely ruined workpiece but an unbroken tap. If you must start the tap by hand, place your hand directly over the hole, straddle the tap handle, and exert moderate downward pressure for the first couple of turns.

Once a tap has started, no further downward pressure is required—so don't use any! The tap will draw itself into the hole as you turn it. Use a real tap handle, not a crescent wrench, and turn it with equal pressure on each side of the tap. Stop the tap every ¼ to ½ turn and reverse the direction to clear the chips. If the tap feels notchy, blow out the chips with compressed air. If it still feels notchy, back out the tap all the way and blow out the chips.

Use the right lubricant. I use Rapid Tap or Tap Magic for every material except aluminum, and Alumitap for aluminum. Any thread-cutting or sulphur-based cutting oil will work on steel and cast iron, as will engine oil (but not nearly as well). Kerosene or diesel fuel works well on aluminum. It is better to tap dry than with the wrong lube. WD–40 does *not* work as a thread-cutting lubricant.

Threaded inserts

Racers and airplane people utilize a lot of threaded holes in castings—particularly nonferrous castings. Murphy has a field day with threads in either aluminum or magnesium. Consequently, a properly threaded hole in either material requires some sort of threaded insert. This is particularly true if whatever is threaded into the hole will be removed from time to time. Otherwise, the threads are guaranteed to strip and, of course, to do so at the most inopportune time imaginable. There are a large number of threaded inserts on the market. To my knowledge, all of them are good. For most applications I use helicoils.

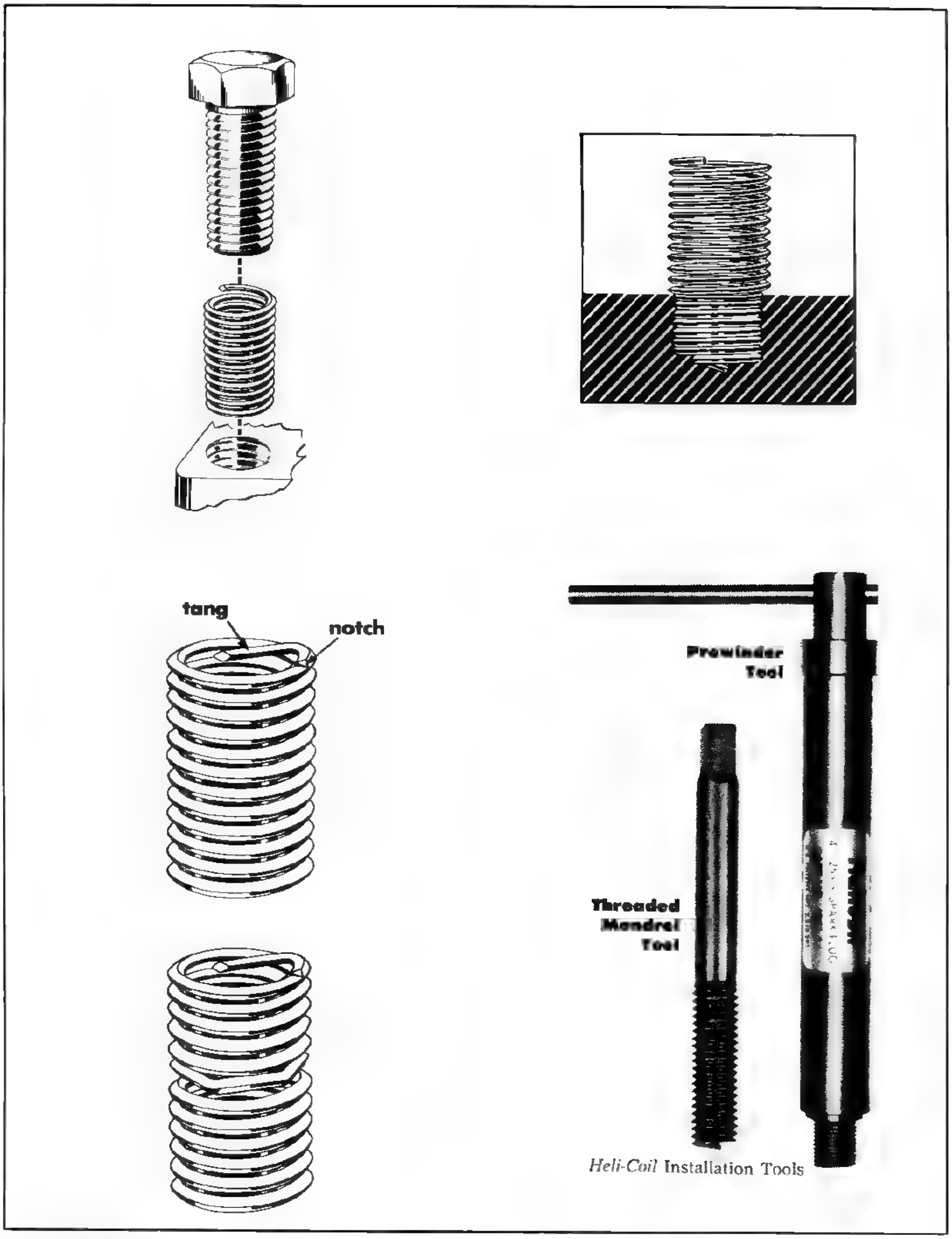
The helicoil

Helicoils are formed of 18–8 stainless steel diamond-shaped wire and designed for the permanent repair of damaged threads. When they are installed in specially tapped holes, helicoils provide permanent female screw threads. They are driven into their holes by driving tangs. The tangs are notched for easy removal after installation.

Helicoils are available in any thread form and pitch that you can come up with and in several lengths. They are also available with a thread locking feature—a series of straight chords formed on one or more insert threads. The male thread elastically distorts the chords into arches and the installed stress locks the wire against the male threads in much the same fashion as the displaced threads of an elastic stop nut. In many cases, oversize helicoils are available so that you get a third chance. After you screw up the original thread, you install a helicoil; after you screw up the helicoil and the helicoil-tapped hole while removing the damaged insert, you retap and use an oversize helicoil. After you screw up the oversized helicoil, you are on your own.

The secret to the helicoil is that, prior to installation, the insert is larger in diameter than the tapped hole. During installation the tool shown here tightens the insert, continuously reducing the diameter of the leading thread so that it can enter the tapped hole. Until the driving tang is broken off, the leading edge of the helicoil presents a smoothly curved surface (like the front of a sleigh runner) to the tapped threads, so the insert proceeds down the helix with no tendency to dig in. After installation, each coil expands outward against the tapped hole to permanently anchor the insert. When the driving tang is broken off, the resulting sharp wire leading edge digs into the tapped threads to prevent further entry of the helicoil during installation of the bolt or stud. The sharp trailing edge of the wire helps prevent the helicoil from unwinding when the bolt or stud is removed.

The first rule of successful helicoil installation is to use the specified tap drill size and the correct helicoil tap. The second rule is to use Helicoil Corporation's own insertion tool. The third secret is



The Helicoil thread insert.

that the coil must be inserted well below the surface of the hole—the sharp end of the coil bites into the tapped thread to provide part of the coil retention. Of course this practice also results in an effectively countersunk hole, which will both prevent raising of the parent metal when the fastener is tightened and remove the end thread from the joint interface.

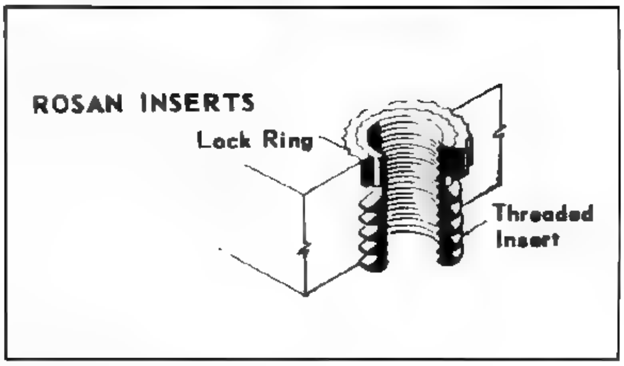
Helicoil removal

A determined racer can screw up anything, and helicoils are no exception. Realizing this, the helicoil people sell a removal tool. I have never seen one. The trick to helicoil removal is to lift the top end of the insert free of the tapped thread, grab it with needle-nose pliers and, exerting just the right amount of tension to reduce the diameter of the insert, unscrew the thing. Ripping it out of its nest will destroy the tapped threads in the parent metal and present you with a serious problem.

Alternatives to the helicoil

I use helicoils, not because I think they are the best inserts available, but because they are universally easy to obtain. The best inserts, in my opinion, are the Rosans. Unfortunately, for all practical purposes, they are unobtainable. My next favorite is Tridair's Keensert. It has the advantage of not requiring special taps and it is positively locked in place against rotation by axial keys. I use them when a crucial bolt is going to be removed frequently.

The Rosan threaded insert.

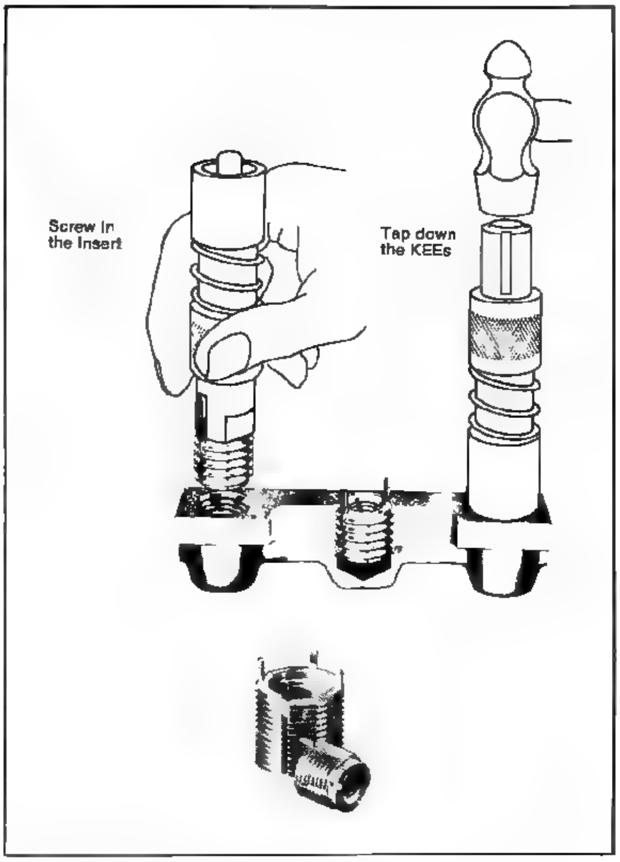


The Rosan threaded insert.

Nuts

A nut is nothing more than a small block of metal (or other material) containing a central hole into whose periphery a female thread has been cut. Not too many decades ago, our grandfathers didn't have a lot of choices in the nut department (of course they had about the same choices in bolts). All of that has changed. Now, thanks mainly to the automotive and aerospace industries and their spin-off technologies, there is choice enough to confuse an experienced fastener engineer. This is both good and bad. Mainly it is good, because we can now find a nut for any conceivable application. It can be bad too, because with so much available it is possible to wind up with a nut that is not suited for a particular application—without even realizing it.

Fortunately, in many of our everyday applications, the actual tensile strength of the nut used in an assembly is of secondary importance to that of



The Keensert threaded insert by Tridair.

the bolt. Notable exceptions are found in engines, airframes and in machine tools. In just about all of our normal applications the assembly will be loaded mainly in shear (hopefully in double shear) and the nut is only required to retain assembled preload/clamping force and not fall off.

As long as the temperature stays below the boiling point of water, just about any properly tightened elastic stop nut will do that job.

Types of nuts

To my way of thinking, there are two basic categories of nuts: plain nuts and self-locking nuts. I will start out by saying that I virtually do *not* use nonself-locking nuts. I am aware that a properly tightened nut should not loosen—either from load or from vibration. But I am also aware that *no one* is going to properly tighten every nut, every time. And even properly tightened fasteners can and do loosen under conditions of severe vibration. More importantly, threaded fasteners *will* loosen when there is relative motion between the parts that they are clamping together, and there is no such thing as a perfectly rigid joint. I look at the elastic stop nut as cheap and very effective insurance. On the premise that if it's not there, no one can put it on, I don't even keep plain nuts in stock.

Having said all that, I must admit that the engine-building fraternity uses plain nuts on cylinder head studs, connecting rod bolts/studs and main bearing bolts/studs. I must further admit that I have never heard of them backing off except when they have been assembled with dirt under the nut. There is a reason. The threaded fasteners related to engines are about the only ones outside the aerospace industry that are properly tightened every time. Even those that are tightened to a torque specification are tightened to a specification determined to be optimum for that particular assembly—not to a value taken from a table. As I said, a properly tightened threaded fastener used in a properly designed assembly will not come loose. . .

Elastic stop nuts

What we generally refer to as self-locking nuts are properly termed elastic stop nuts because they form their lock by controlled elastic deformation of a portion of the thread cylinder. There are three basic types of elastic stop nut: those that use a nylon locking collar; the displaced thread, elliptically offset or triangular displacement all-metal nut; and the slotted beam, segmented or flex lock all-metal nut. The operating principles of all three are similar, and I am going to take the time to explain them individually for the simple reason that no one else seems to.

Nylon collar elastic stop nut

The most common (and least expensive) of the self-locking nuts utilizes a nylon locking collar. It is available either as an AN/MS item or as an industrial part. You can be certain of the material, design and quality control on the AN/MS parts. The inside diameter or I.D. of the nylon insert is slightly smaller than the major diameter of the bolt thread. The nut will spin freely on the bolt until the bolt threads engage the locking collar where they impress (but do not cut) mating threads in the nylon. This compression forces the upper flanks of the nut threads into metal-to-metal contact with the lower flanks of the mating bolt threads, forming a friction hold. This hold, combined with the compression of the nylon, is sufficient to ensure that a properly tightened nut will not loosen on the bolt at temperatures up to 250 degrees Fahrenheit; above that temperature the nylon loses its elasticity.

Nylon collar elastic stop nuts cannot harm the bolt threads, are reuseable many times and are also available as castellated nuts. They are manufactured to AN specs as AN-365 (MS20045) full-height tension nuts and as AN-364 (MS20064) ½ high, or thin, shear nuts. They are also manufactured as commercial units. I use the AN variety.

Nylon pellet nut

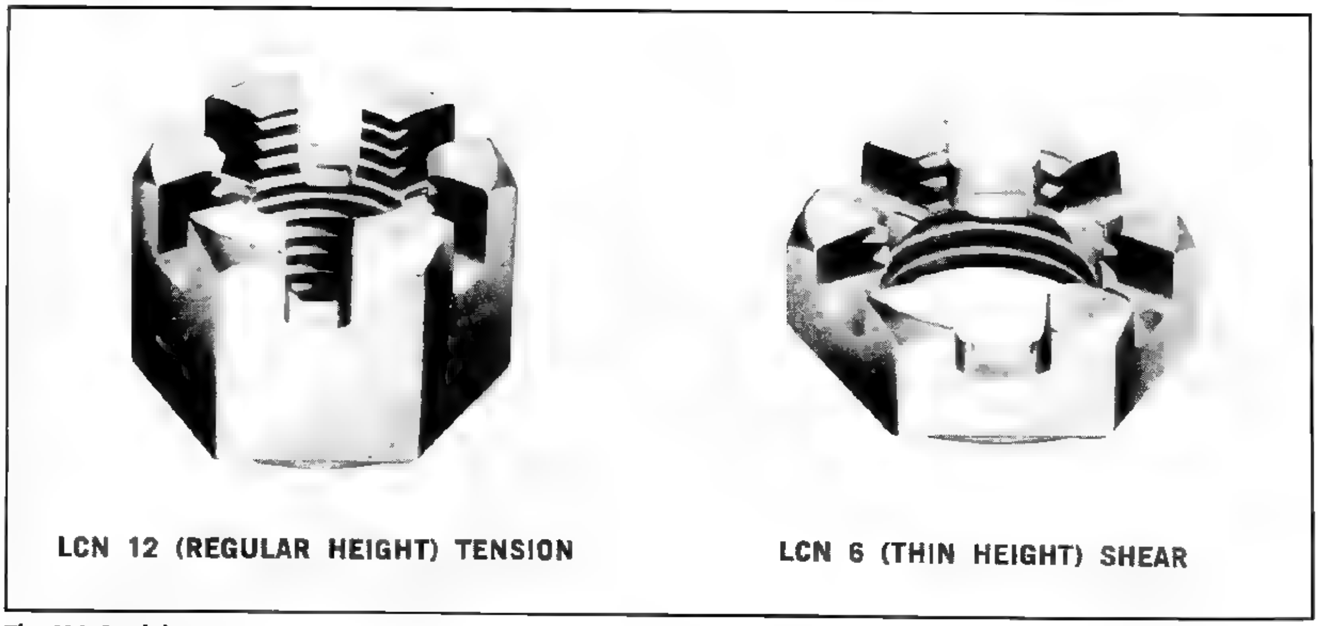
The nylon pellet nut is a lower cost and less effective variation on this theme. The operating principle is the same but since the elastic area is considerably less, the resistance to vibration is greatly reduced. The nylon pellet nut is suitable only for applications that will not be subjected to severe vibration. While they are fairly popular in production use, I see no reason for people like us to ever use a pellet nut.

Elliptically offset elastic stop nut

The threads of these nuts are divided into two separate sections. The lower section of threads are manufactured in the form of the normal cylindrical helix. The upper portion of the thread is distorted or offset into either a triangular or an elliptical shape. As the male thread enters the distorted locking section of the nut thread, the out of roundness is elastically displaced to a cylindrical section. This displacement is actually elastic deformation of the nut material. When the male thread protrudes two or three threads beyond the top of the nut, the elastic deformation has created a friction hold sufficient to lock the nut against minor vibration—even though it may not yet be seated against the work surface.

When it is seated and fully tightened, the base of the nut presses down on the clamped surface while the locking threads press in against the bolt. This action lifts the nut and forces the upper flanks of the retaining threads hard against the lower flanks of the bolt threads. The fastener is now fully locked. This forms a stronger and more temperature-resistant lock than the nylon collar.

These nuts are available in tensile strengths to 300,000 psi with temperature ranges up to 1400 degrees Fahrenheit. When manufactured to mil-



The SPS Castleloc nut.

itary specs they are almost indefinitely reuseable and will not harm the male threads of the bolts that they are meant to be used with—at least for many applications. However, if you use a 220,000 psi nut on a 125,000 psi AN bolt, then the hard deformed threads of the nut are going to act like a misshaped thread die on the relatively soft threads of the bolt and you will lose the bolt threads. On the other hand, some of the industrial copies are not meant to be disassembled and can wreck male threads in a single application.

Slotted beam elastic stop nut

The thread of the all-metal slotted beam (or segmented) elastic stop nut is also divided into two separate areas—the retaining threads and the locking threads. The upper locking threads are radially slotted and deflected inward to produce a controlled friction lock at the top of the nut. Again they are available in various configurations, alloys and tempers to about 180,000 psi and 1400 degrees Fahrenheit. When manufactured to military specs, they are reuseable at least a dozen times and are harmless to bolt threads.

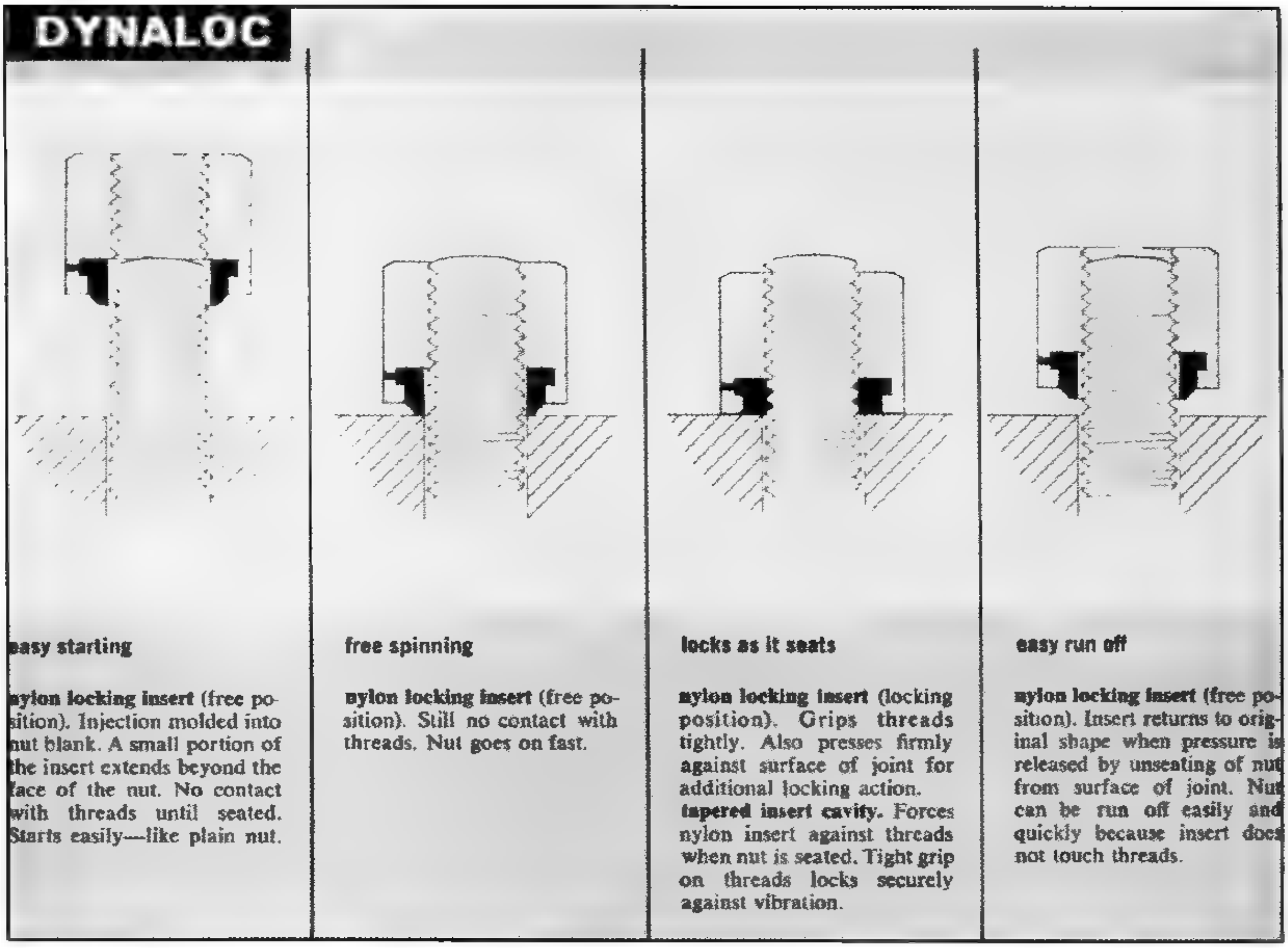
An interesting variation is SPS's Castleloc lock nut. There are some super-critical applications where the designers want a fail-safe back-up if a cotter pin should fail or be left off. The Castleloc is a castellated nut in which the threaded segments between the castellations are deflected inward to produce a friction lock.

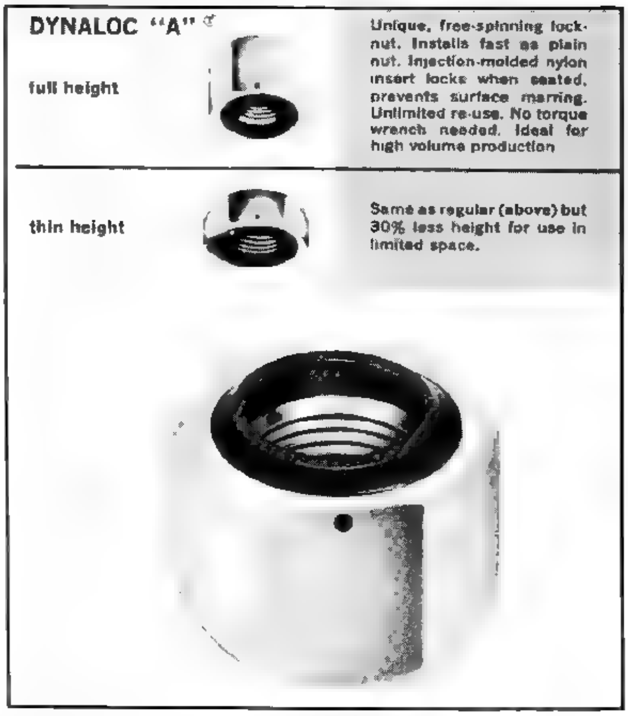
Free-spinning elastic stop nuts

The standard types of elastic stop nuts present some problems in production and in maintenance. One of the chief disadvantages to their use is that they are not free spinning and so take longer to install than plain nuts. At least two types of free-spinning self-locking nuts have been developed to alleviate this situation: the captive washer lock nut and the Dynaloc free-spinning elastic stop nut.

Any number of companies manufacture plain nuts with a captive toothed or serrated lock washer attached to the base of the nut. These are every bit as effective as the two separate items and have the advantage of ensuring that the lock washer will not be left off the assembly. They also simplify inventory control. Unfortunately, as you will find out

Captive washer lock nuts.





The SPS Dynaloc free-spinning elastic stop nut.

when we discuss thread locking devices, I don't think that the lock washer is a particularly effective device to start with.

SPS's Dynaloc uses a nylon ring molded into the bearing face of the nut. The inside diameter of the ring is slightly greater than the thread diameter, and it projects downward from the face of the nut. In use, the Dynaloc nut spins freely onto the male thread—until the locking ring contacts the work face. At this point the nylon is displaced into the thread and forms a lock similar to that of the standard nylon collar self-locking nut. These devices are considerably superior in vibration resistance to the star washer type. It is well to remember that they will not work at all if they are installed like a standard nylon collar nut, with the locking ring up.

High-tech options

The aerospace industry, accustomed to virtually unlimited budgets and concerned not only with vibration but also with weight and very high and very low temperatures, tends to use the MS21042 series of Jet nuts or K nuts for just about

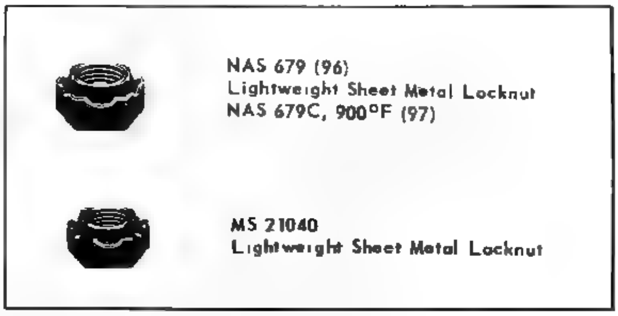
everything. So do the Formula One and Indy car racers. These are fine pieces indeed—elliptically offset, light, temperature resistant, positive locking, handsome and almost indefinitely reuseable. They are also diabolically expensive and their haphazard use smacks, to me, of conspicuous consumption. I use plain old nylon collar elastic stop nuts almost everywhere. I have never had one back off. For reasons that have more to do with faith and quality control than they do with strength, I use the AN/MS stuff rather than the industrial variety.

Practical weight savings and other concerns

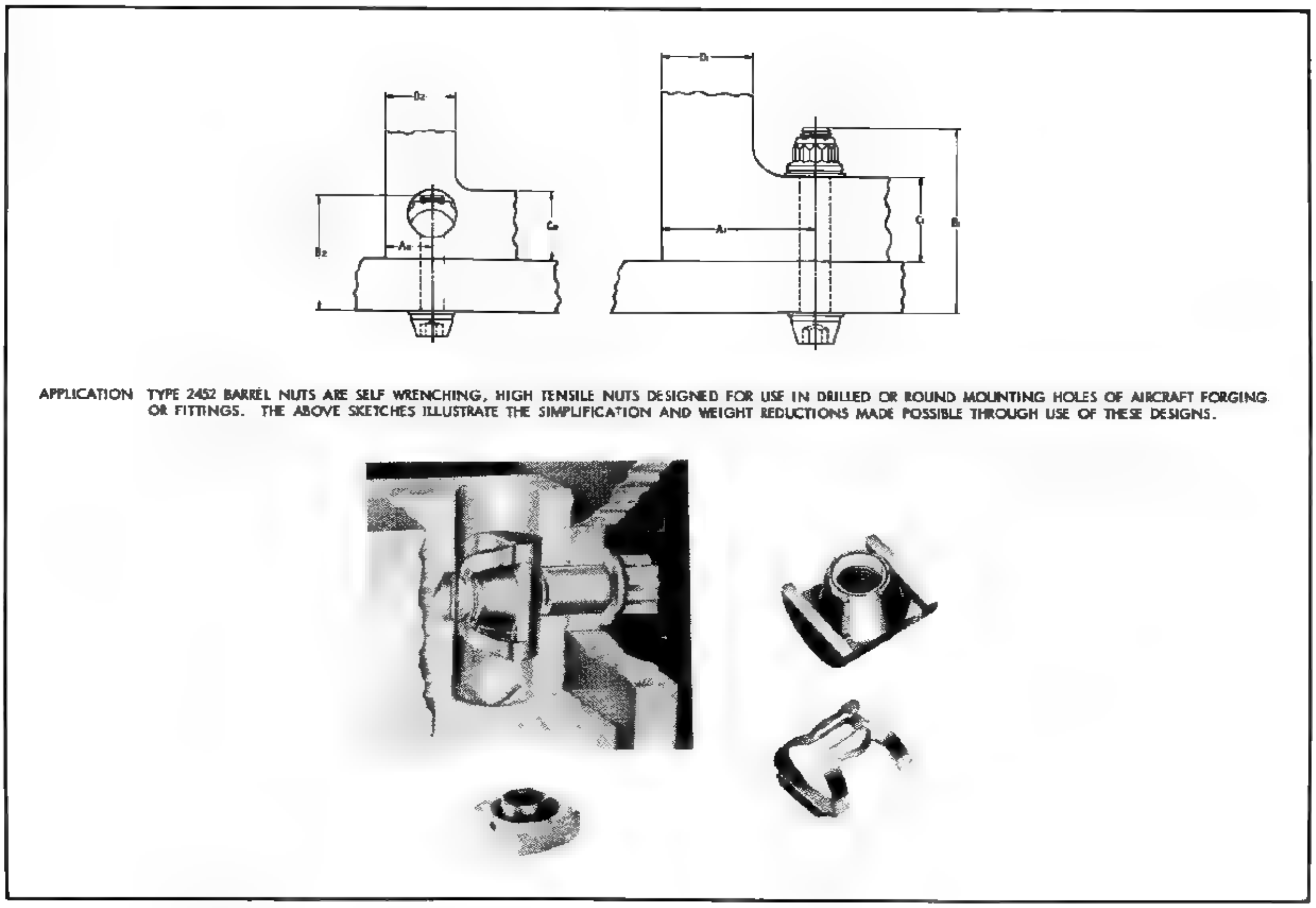
When it comes to race cars, I am at least as weight conscious as the next person. So I use shear (½ high) nuts on double shear applications. I also use shear bolts when I can find them on the surplus market, and I trim the excess thread length from standard airframe bolts when I use them in double shear. I consider single shear, a mortal sin to begin with, to be a tension application so far as fasteners are concerned.

For high temperature, high stress and/or severe vibration applications, I use all-metal six-

K or Jet nuts.



NAS–679 lightweight elastic stop nut.



The barrel nut alternative to heavy bolting flanges.

point jet nuts. If I were really in the bucks, I would use the NAS-679 series high-temperature lock nuts which are formed from sheet metal and are really light. They also have the intriguing feature that they can be either external or internal wrenching (although a special internal wrenching socket is required). If you are talking about turbocharger-type heat, you should talk to either the manufacturer of the turbo or to SPS.

Nylon insert elastic stop nuts should not be used on modular wheels. They simply will not stand up to the heat generated by the brakes. Bill Jongbloed, who manufactures what I consider to be the best modular wheels in the world, uses an all-metal lock nut manufactured by Fastron that is light, has a lot of bearing area, holds its torque indefinitely while in very close proximity to the brake disc and is relatively inexpensive. It also tends to chew up the male threads when reused. Jongbloed does not recommend reusing the nuts. Being no fool, he uses SPS Unbrako cap screws to hold his wheels together. I do not reuse the cap screws either.

Barrel nut

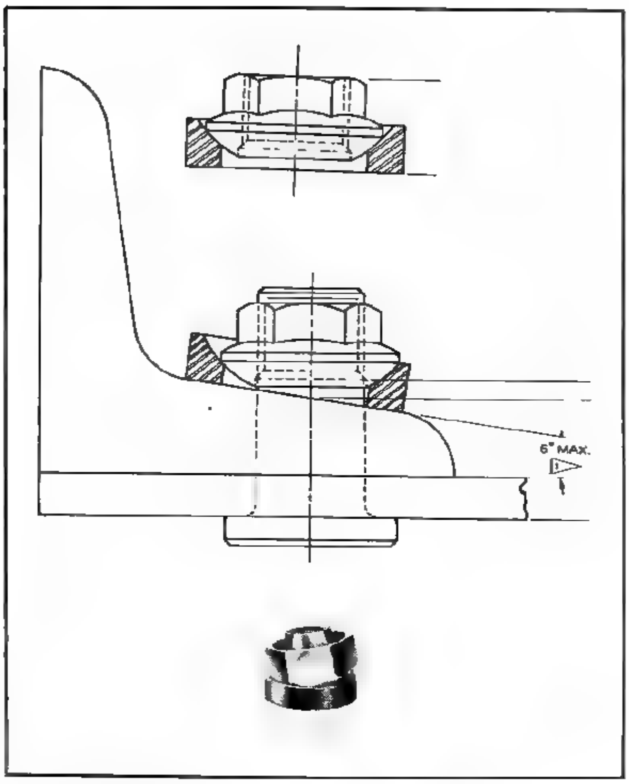
Generations of engineers have detested bolt flanges. They are heavy and they are bulky and they do nothing but allow thin-walled components to be bolted together. The aerospace industry cannot afford much nonproductive bulk, let alone weight. The barrel nut was developed to reduce the amount of redundant mass required to bolt thin sections together. All that is required to use a barrel nut is a round access hole. Barrel nuts are captive or self-wrenching and are available in a multitude of fixed and floating (caged) styles in all the normal materials and locking configurations.

Self-aligning nut

Earlier I pointed out the importance of making sure that nuts and bolts are always installed square to the work surfaces. Sometimes this is not as easy as it sounds, particularly when working with tapered aircraft extrusions. The traditional method of squaring the bearing surface of the work involves spot facing the hole, which is a time consuming pain. Self-aligning nuts and washers are available from (among others) SPS and ESNA. Like the rest of the specialized hardware, they are not easy to find unless your name is Douglas, Boeing or Northrup and you need a few tens of thousands of the things. I mention them not because any of us are going to order them, but to point out that there is a fastener to solve just about any problem that anyone can come up with. As a point of interest, the construction industry runs into the same problem; cast iron square washers are available through your local industrial hardware supply house.

Plate nut

One of the most practical spin-offs from the aircraft industry is the nut plate—also called the plate nut. This clever device provides a quick and relatively inexpensive way to end up with a captive



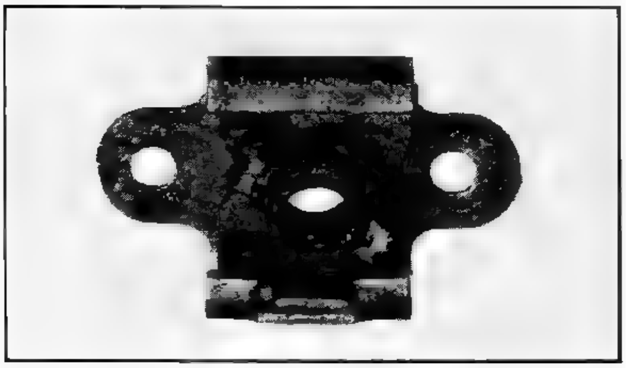
SPS self-aligning nut and washer assembly.

The basic plate nut.

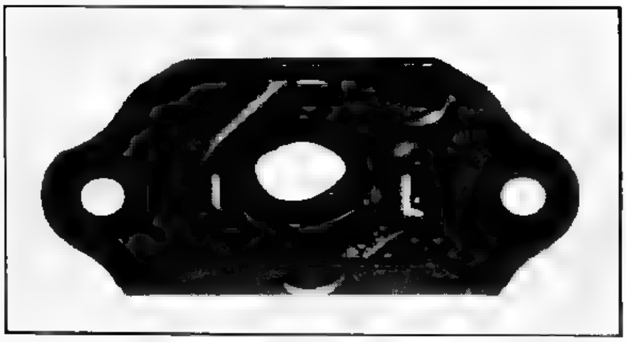
self-locking nut wherever you might need one. They are available in a seemingly endless variety of shapes and configurations. The aircraft industry uses them on the backside of sheet-metal panels. I use them anywhere that I figure I will have trouble holding a nut when I am in a hurry. I used to be more selective about where I put them because they are a pain to replace when the locking element wears out. Mike Torino at Torino Motor Racing, Ltd. has found the answer: the nut plate with removable elastic stop nut—I now use them without reservation.

In order to ensure that everything lines up and fits when the job is done, some care is necessary in the layout and installation of nut plates. Nut plate drill guides are available at nominal cost from your local aircraft supply house, or you can make your own. If you are going to install several nut plates, the drill jig will pay for itself quickly.

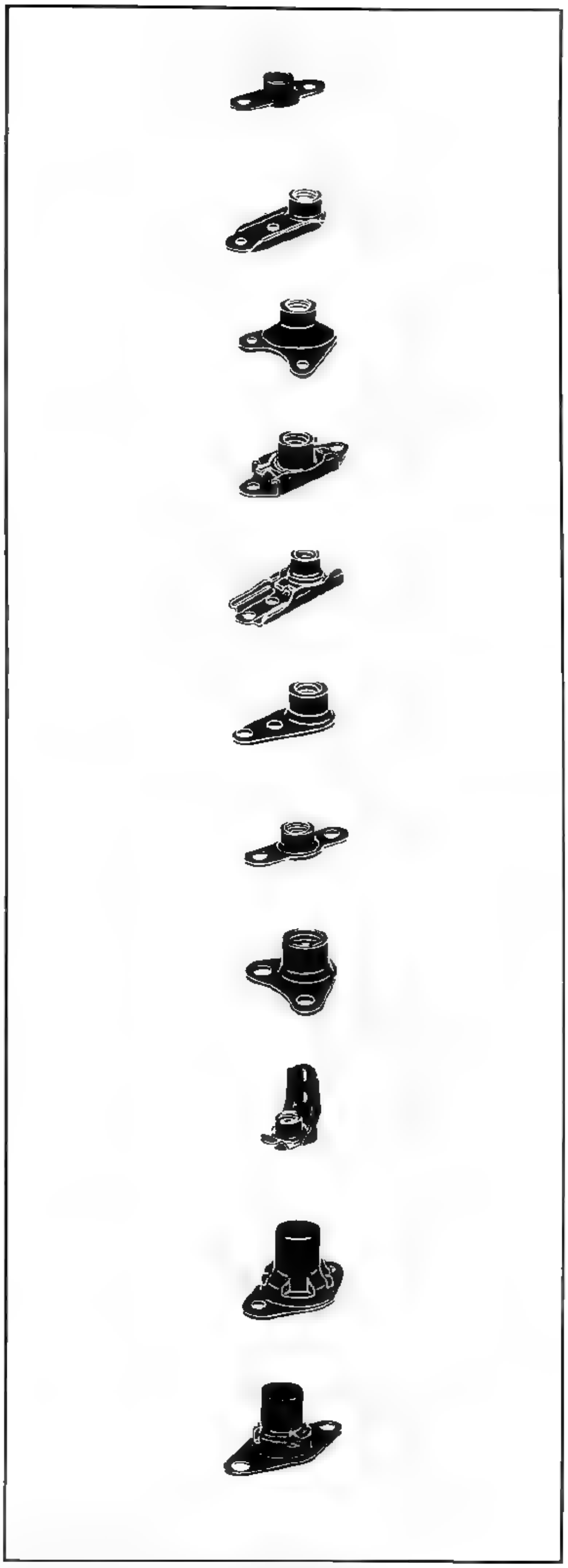
Racers being the way that we are, most times you get to do it without the tooling. The usual method is to drill and deburr the machine screw holes in the item to be mounted and then carefully transfer the holes onto the panel that will receive the nut plates. The nut plates are then temporarily attached to the panel with short machine screws and used as drill guides for the rivet holes in the panel. After more years than I care to think about, I



The floating plate nut.



Caged plate nut with replaceable nut from Torino Racing.



Some available plate nut configurations.

have finally figured out that it is a damned sight more practical to carry around a bunch of 3/32 in. blind rivets than it is to drill out the 3/32 in. holes that come in #10–32 nut plates to 1/8 in., so that you can use standard pop rivets.

Before installing the nut plates for real, it is necessary to deburr and perhaps countersink the rivet holes. To make sure that everything will fit, during the deburring process I drill the machine screw holes 1/16 in. oversize. Nut plates are available with the nut part floating in a cage so that some alignment can take place after installation. They are also available as edge clip mounting units. This does away with most of the installation procedure—just drill a hole for the machine screw and go. I sometimes use the edge clips to mount the Gurney extensions on wings.

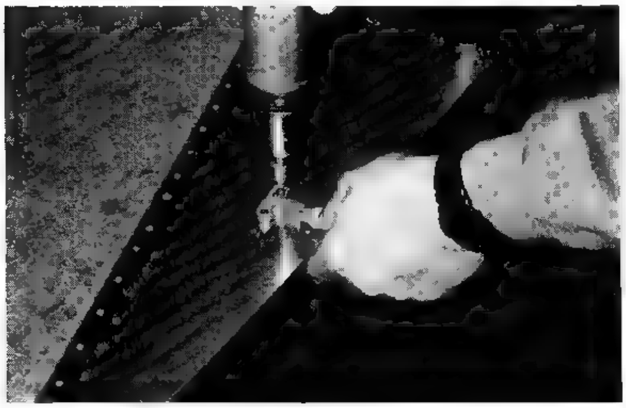


Plate nut drilling jig.

Internal wrenching nuts.

Be careful with surplus nut plates. Some are old enough to feature the fiber locking collars which preceded the use of nylon. These are easily detectable—the fiber is dull, usually red in color and looks like compressed paper. It will crumble if compressed with a knife point. The nylon can be any color, but it is always shiny in appearance and will not crumble. They are only good for about a half dozen uses before replacement is necessary. Some *look* like all-metal elastic stop nuts, but function by collapsing an open thread and are a one-time only lock. I buy my nut plates from Earl's Performance Products, Torino Motor Racing or Aircraft Spruce.

Internal wrenching nuts

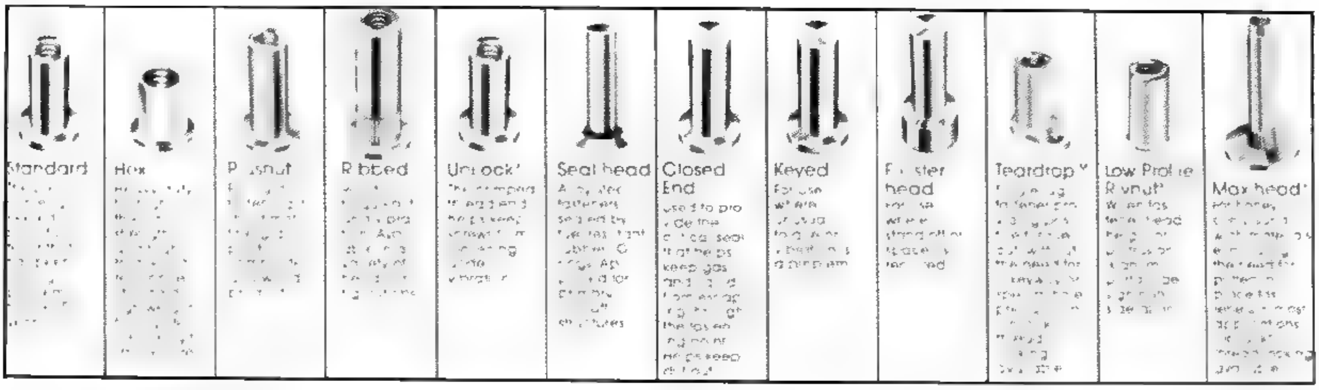
Because I am one, I can state with some authority that engineers are forever designing themselves into corners. One of our favorite little tricks is to end up with insufficient room to apply a wrench to either the nut or the bolt. When the offending item is the bolt, we usually end up using an internal wrenching bolt. When the problem is with the nut, we have some choices. Sometimes we can use a plate nut of one sort or another. Other times we can redesign and use a barrel nut. Very often the problem can be easily solved by the use of an internal wrenching nut.

The same clever devils who came up with and gave their name to the internal wrenching bolt, the Allen Manufacturing Company, make a matching high-strength nut. This gadget has saved me several times, since I first discovered it during a period of desperation in the late 1960s. They are available with either six-point or twelve-point wrenching sockets and as elastic stop nuts. Good machine shop supply houses stock them.

Clinching nut

The plate nut meets most of the requirements for captive self-locking nuts but they are time consuming to install, particularly in production. Furthermore, you must have access to both sides of the work in order to install them. Clinch nuts and swadging nuts are one way to wind up with load bearing female threads in sheet metal and in thin-sectioned castings or forgings when you have access to only one side of the work. Clinch nuts work like pop rivets. They are installed through predrilled holes and compressed with a pulling tool. When compressed, the unthreaded portion of the nut bulbs out and grips the sheet metal. Special tooling is required and the nut must be closely matched to the metal thickness.

Several companies manufacture clinch nuts, and the tooling is usually not interchangeable between brands. This is one of those cases where it pays to read and follow the instructions closely. If the nuts are not installed exactly according to instructions, they will probably allow the initial installation of the matching machine screw, but will



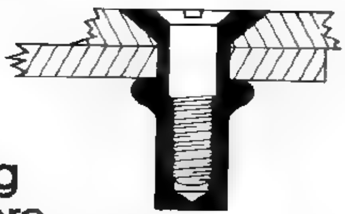
The B. F. Goodrich Rivnut.

turn in the metal section during removal—presenting the user with a distinct problem.

Rivnut

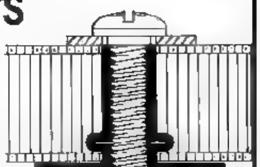
The grandfather of the clinch/swadge nut is the Rivnut. Sometime in the 1930s, B. F. Goodrich developed the rubber de-icing boot for aircraft wings. Now virtually forgotten, the boot made all-weather flying possible in the days before jets. (Jets fly over the weather and are not subject to icing.) The de-icing boot was formed like a long balloon and was installed over the leading edge of the wing. When ice began to form, the boot was activated

Aircraft Fuel Tanks



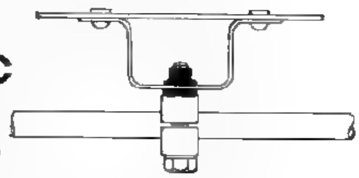
The "O" ring seal on a precision Seal-Head RIVNUT fastener provides for a crucial liquid tight rivet.

Aircraft Interiors



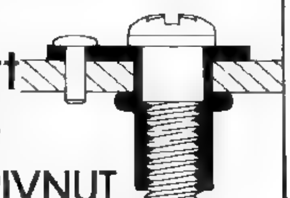
The Maxihead™ RIVNUT with the UNILOCK® thread locking feature secures aircraft galley equipment to honeycomb/sandwich walls, eliminating costly potted-in-place fasteners.

Aircraft Hydraulic Lines



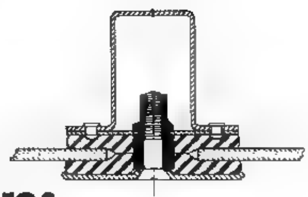
The RIVNUT fastener anchors the hydraulic line support in an aircraft while weighing less and costing less than an anchor nut.

Aircraft fuel-cleanout port access holes



The Teardrop™ RIVNUT fastener with the UNILOCK® thread locking feature is designed for high-torque anchor nut fastening applications.

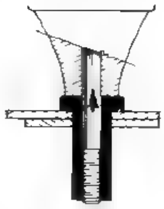
Helicopter Window Panels



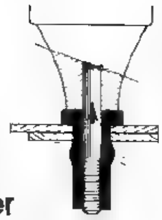
The RIVNUT fastener secures plastic window panel to the frame and forms seal designed to withstand vibrations and weather.

Step 1
The RIVNUT fastener is threaded onto the mandrel of an installation tool.

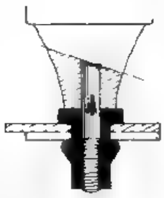
Step 2
The RIVNUT fastener, on the tool mandrel, is inserted into the hole drilled for installation.



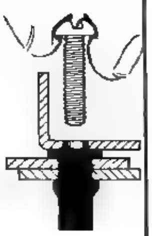
Step 3
The mandrel retracts and pulls the threaded portion of the RIVNUT fastener shank toward the blind side of the work, forming a bulge in the unthreaded shank area.



Step 4
The RIVNUT fastener is clinched securely in place. The mandrel is unthreaded, leaving the internal RIVNUT threads intact.



Blind Nut Plate
The final result is a strong, secure blind nut plate for simple screw attachments.



with compressed air. A regulator forced the boot to pulsate and break off the ice before it was thick enough to destroy the lift of the wing. A generation of pre-jet pilots thanked God for B. F. Goodrich every night of their lives.

In order to make the de-icing boot practical, B. F. Goodrich had to figure out a way to install the things—and to remove them when they needed to be replaced. So a rubber company developed the first clinch nut. The Rivnut is an internally threaded blind rivet. Easily installed with what used to be relatively inexpensive tooling, it provides a stable female threaded anchor in thin sheet metal at reasonable cost. Avdell and others make similar widgets, but I prefer the Rivnut because it is not particularly fussy about exact hole size and because, if it has been properly installed, it is almost impossible to twist out.

It is available in protruding or countersunk head styles, sealed or open, in both steel and aluminum. B. F. Goodrich's patent must have run out sometime in the past few years, as a lot of people now market what appear to be exact copies. The copies are considerably less expensive and, as near as I can tell, they are every bit as good.

Nut-Sert

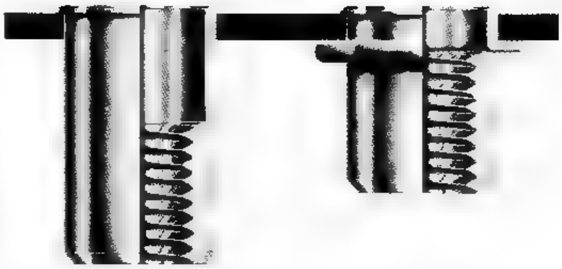
The Avdell Corporation manufactures a clinch nut called the Nut-Sert. I have never been able to make Nut-Serts work very well. Even when I ream the holes to the specified size, they are prone to spinning when I try to remove the machine screws. I don't use Nut-Serts or their imitations.

Swadge nut

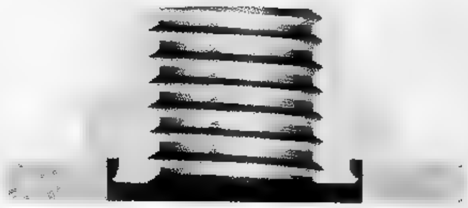
The swadge nut is similar in result to the clinch nut, but it arrives at its result by a different path. The clinch nut deforms and grips the sheet metal between its manufactured head and the shop-formed bulge. The swadge nut displaces the sheet metal to which it is mounted into serrations in its body. The swadge nut is basically an aerospace and high-production unit. The tooling is expensive and the directions must be followed exactly. To my knowledge, no one uses them on high-performance vehicles outside of aerospace.

Well-Nut

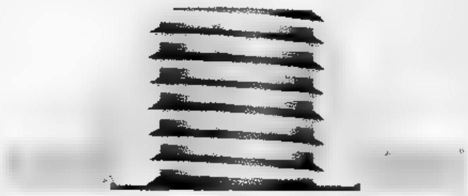
One of the more clever and useful female threaded inserts is the Well-Nut by United Shoe Machinery (the same people who gave us the pop rivet). The Well-Nut is a flanged neoprene bushing with a brass nut cast into its body. It can be inserted



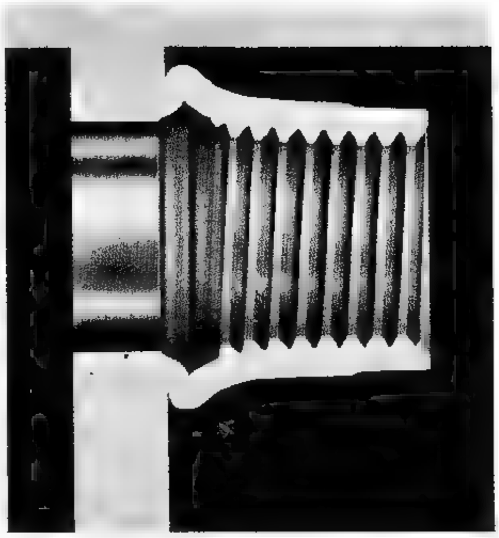
The Avdell Nut-Sert



#8-32 Nut inserted in hole ready for swaging into plate. Cross-section enlarged four times.

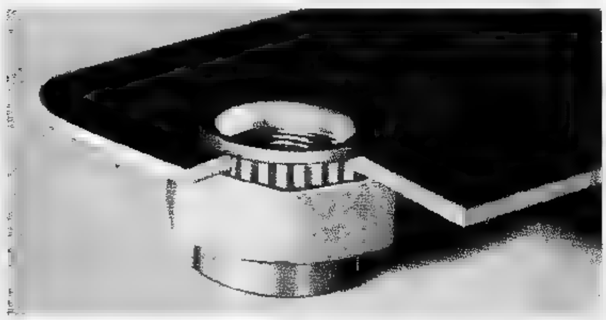


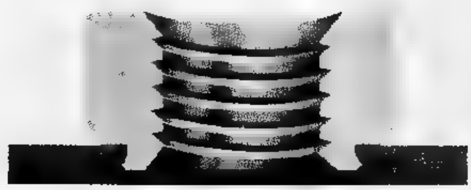
Cross-section of #8-32 Nut after installation. Note positive flow of metal into groove in nut, imbedded knurling ring and smooth faying surface on bottom of plate.



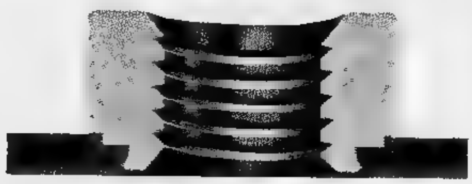
BF Goodrich Rivnut.

4 As the shank of the Clinch Nut is forced through the sheet the shank is peened over.



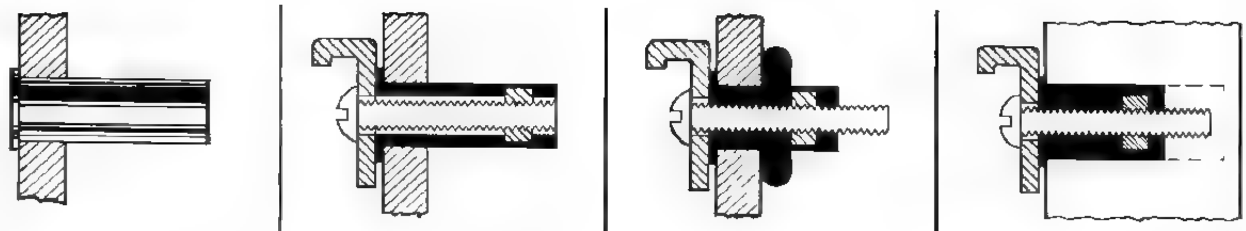


#8-32 Nut inserted in hole ready for swaging into plate. Cross-section enlarged four times.



Cross-section of #8-32 Nut after installation. Note positive flow of metal into groove in nut, imbedded knurling ring and smooth faying surface on bottom of plate.

Swadge nuts.



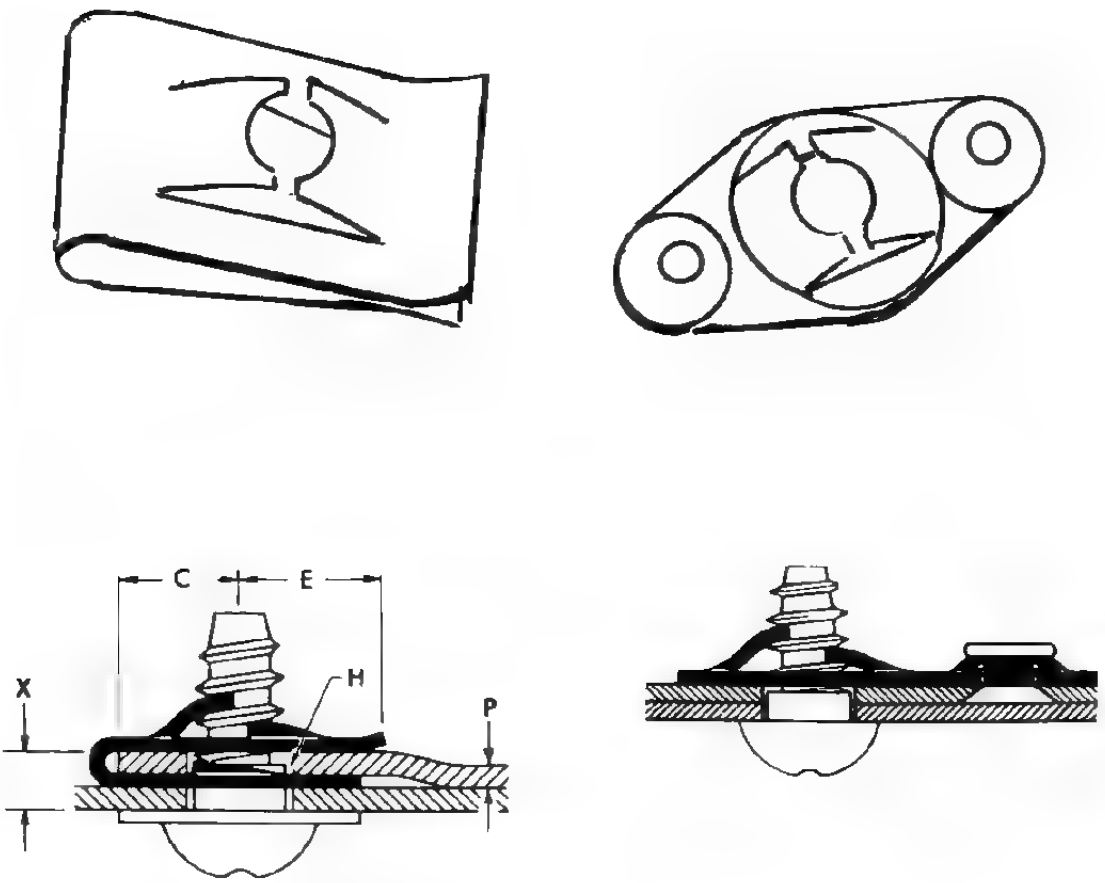
1. Well-Nut is inserted in a pre-drilled hole, with its flange against the outer surface. Since Well-Nut is a blind fastener, it is installed from one side of the work with no need for access to the inner side.

2. The part to be assembled to the surface is placed against the flange of the Well-Nut and is secured by a machine screw engaging the captive brass nut.

3. As the machine screw is tightened, the neoprene body of the Well-Nut is compressed and expanded, forcing it tightly into the screw threads and against the inner surface of the thin-wall material.

4. Installed in a blind hole in solid material, the body of the Well-Nut expands tightly against the walls of the hole, effecting a secure, dependable fastening.

The Well-Nut.



The Tinnerman nuts.

either through a hole in a sheet panel or into a hole in solid material. In either case, a machine screw is screwed into the brass nut. In the panel installation, tightening the screw compresses and expands the neoprene. This forms a bulb on the far side of the panel similar to that of a riv-nut and firmly attaches the Well-Nut to the panel. At the same time the neoprene tightly grips the threads of the machine screw and effectively locks the assembly.

When a machine screw is turned into a Well-Nut that has been installed in a hole drilled in solid material, the neoprene expands against the walls of the hole, tightly gripping the assembly.

The Well-Nut is not a particularly strong insert, but it has some unique features: first, it dampens vibration and absorbs shock; second, it seals to the material against air or liquid leakage; and third, it can be installed either in thin walls or in solid material.

The vibration dampening and shock absorbing qualities make the Well-Nut ideal for shock mounting panels and vibration-sensitive components such as electronic ignition modules, fuel pumps and so on. In the days of aluminum fuel tanks, we used to weld tubes into the tanks themselves to serve as Well-Nut recesses in order to isolate the tanks from vibration.

The one thing that United Shoe Machinery or USM doesn't tell you in their Well-Nut brochure is that both the inner and outer edges of the mounting holes in thin panels, and the outer edge of a hole in solid material must be completely deburred and slightly radiused before a Well-Nut is inserted. When tightened against a sharp edge, the Well-Nut circumcises itself. Polishing a radius is easy enough on thick material, but on thin panels it may be necessary to polish only the inner edge and to use a radiused washer on the outside of the hole.

Tinnerman nuts

The Tinnerman nuts (speed nuts and sheet-metal nut) pictured are clever and useful devices. They are self-locking and come in every configuration imaginable. Meant for use with sheet-metal screws, Tinnerman nuts are formed from heat treated steel-sheet. They feature a pair of opposed, arched prongs through which the screw threads. As the screw is tightened, the prongs are displaced inward and lock against the root of the screw thread. Final tightening flattens the arched base and preloads the nut so that a constant tension lock is applied to the screw threads. This is all very well, but as I pointed out earlier, I don't use sheet-metal screws . . . so I don't use Tinnerman clips either.

Variety

It is not possible in a book of this length to discuss the properties and uses of the specific nuts that are available. If you believe that variety is the spice of life, you will love the world of nuts. The variety of different types and configurations of nuts and inserts currently available to the aerospace industry is staggering. I think that I have covered the basic types that are of interest to those of us who work with high-performance machinery. The appendices list some of the catalogs available to those of you who want to go into more detail.

7

Locking devices

The most effective and reliable method of preventing any nut or bolt from loosening is to tighten the thing properly to start with. When a threaded fastener is properly tightened, the residual stress within the bolt will lock the assembly. As I described in chapter three, the bolt's attempt to return to its original length wedges the top surface of the male threads against the bottom surface of the female threads, creating a very effective lock.

Even the best mechanical locking devices can only supplement this action, they cannot replace it. The next most effective method, where the application allows its use, is the elastic stop nut. Regardless of the effectiveness of residual stress as a locking medium, for those applications where the fastener will be subjected to high levels of vibration, cyclic stress and/or turning torque, common sense, the survival instinct and the FAA require some sort of additional positive antirotation or locking device. In critical applications these devices also act as insurance against human error.

Lock washers

There are a great many locking devices on the market today. The most popular is also the least effective—the lock washer. There are three basic types of lock washers: the spring washer, the wave washer and the serrated, or star, washer.

Neither the spring washer nor the wave washer do anything worth talking about—other than to provide the user with a false sense of security. Think about it for a moment. From experience, you know that it takes very little load to compress a spring washer. For example, the spring washer will be completely closed long before we reach recommended torque when tightening a bolt. Once compressed, the spring washer is nothing but a flat washer. If, for whatever reason, a bolt should loosen to the point where the spring washer opens enough to become a spring, there was too little residual stress in the assembled bolt for any sort of safety. In other words, the thing wasn't tightened sufficiently. Exactly the same is true of the wave washer which is, for some reason or other, popular in Germany. If you decide to use a spring lock washer, a flat washer should be placed between the lock washer and the work surface to prevent damage to the surface. This is not necessary with the wave washer.

I am willing to admit that there are installations where the serrated or star washer can be effective. These installations are limited to the smaller sizes and almost always have to do with machine screws bearing on a relatively soft surface—aluminum or plastic, for example. The teeth of the washer can and do bite into the surfaces of soft materials and offer reasonably positive protection against rotation. They are available with either internal or external teeth, and also as coned washers for countersunk bolts.

I try not to use lock washers. I use prevailing torque-type self-locking nuts on all through holes, and check or jam nuts to lock rod end bearings and threaded adjusters. With blind holes, if I do not trust the thread tension of a properly tightened bolt, I use the appropriate grade of Loctite and/or safety wire. I do, however, carry a selection of aircraft spec (AN-935) spring lock washers around with me—just in case. I will not use industrial spring lock washers because they are liable to be too brittle for my taste.

Tab washer

I consider the tab washer to be an idiot device. In order for us to be able to bend the tab over the face of the bolt, the tab must be relatively soft. In service, cyclic stress may well cause the soft washer to squeeze out sufficiently to allow the bolt to relax its tension. It doesn't take much rotation to lose the residual stress. When this happens, the clamping force is lost and the fatigue life of the bolt itself will be shortened by a remarkable amount. Thus this supposed safety device has become a hazard. I do not allow the use of tab washers on my projects. If you must use a tab washer, use a stainless steel one. Never reuse a tab washer.

Key washer

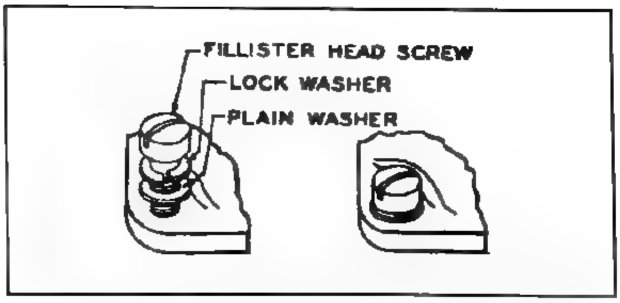
The key washer is nothing more than an internal tab washer. The tab is meant to fit in a slot like a woodruff key, while the washer gets bent and pounded against the flat of a nut. Everything that I just said about the tab washer applies to the key washer, except that I do use them when I have no choice, with engine pullies, vibration dampners and the like—but I don't trust them.

Safety wire

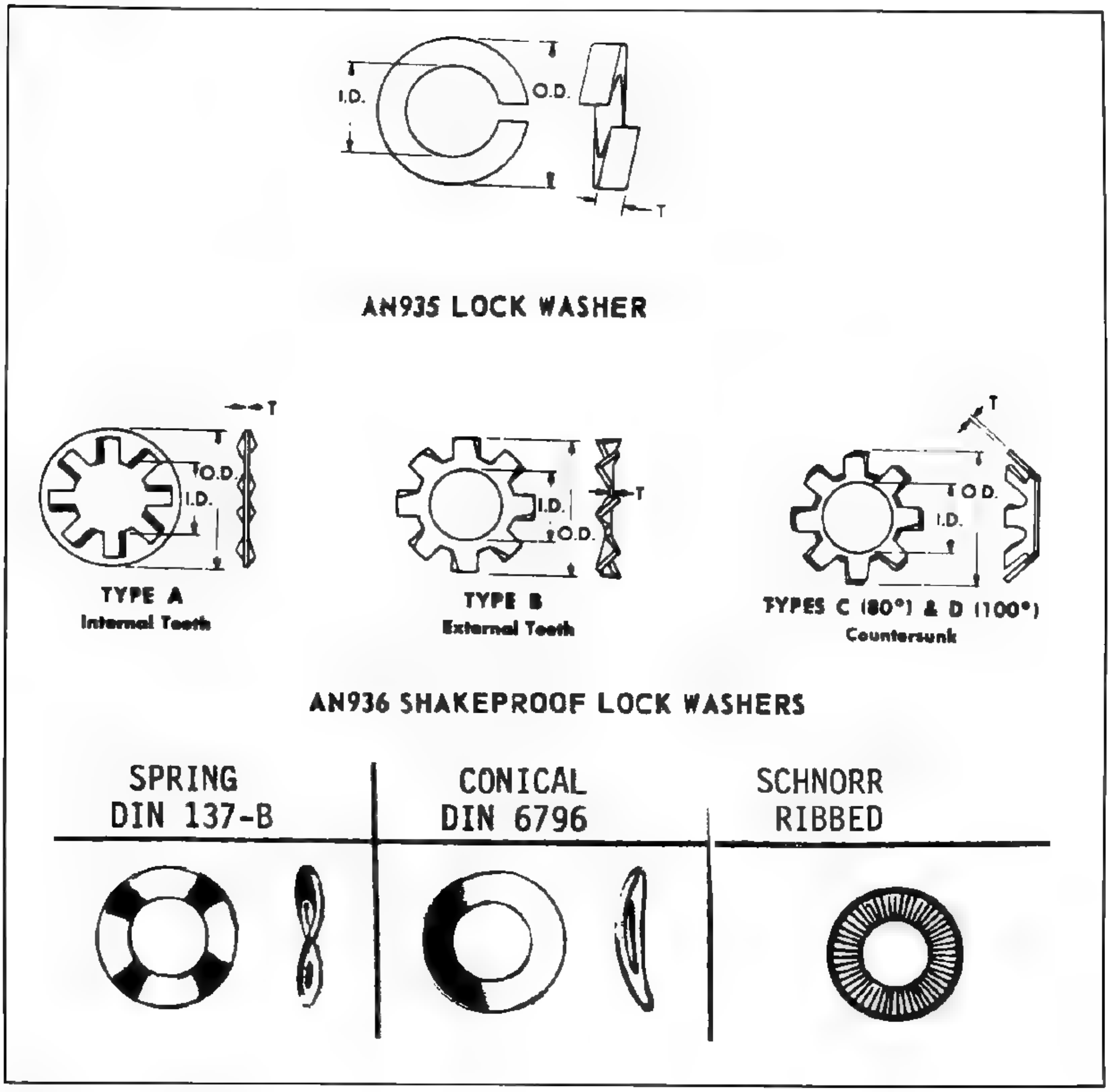
I use a lot of safety wire. I use it for two reasons. First, safety wire on a bolt head provides an

easy and positive visual indication that someone has tightened it. We don't even want to think about the mentality that would allow anyone to safety wire a bolt without first checking to make sure that it is tight.

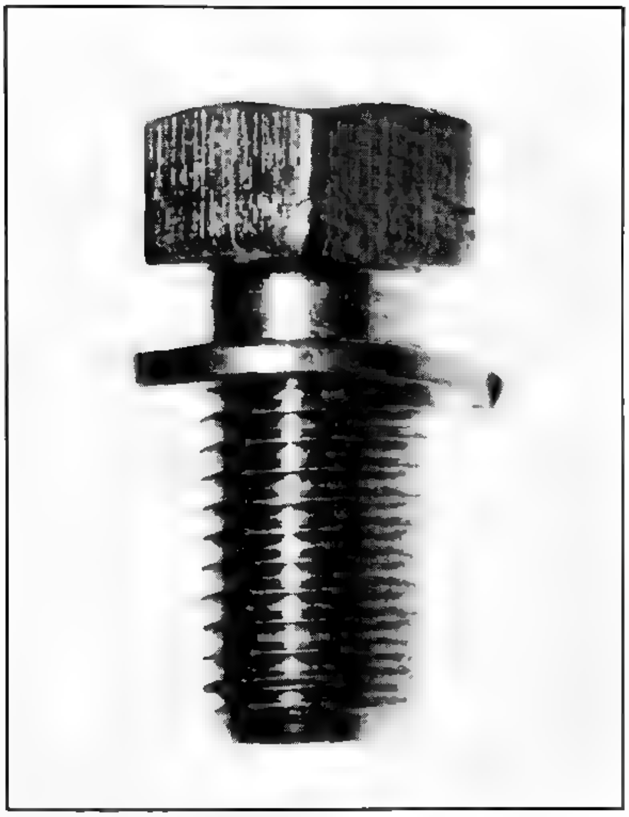
Second, if the bolt does loosen, the safety wire will prevent it from falling out. Contrary to popular belief, even the best job of safety wiring will contribute virtually nothing to the task of preventing a bolt from loosening to the point where effective levels of residual stress disappear. All that the safety wire can do is limit the rotation of a bolt and pre-



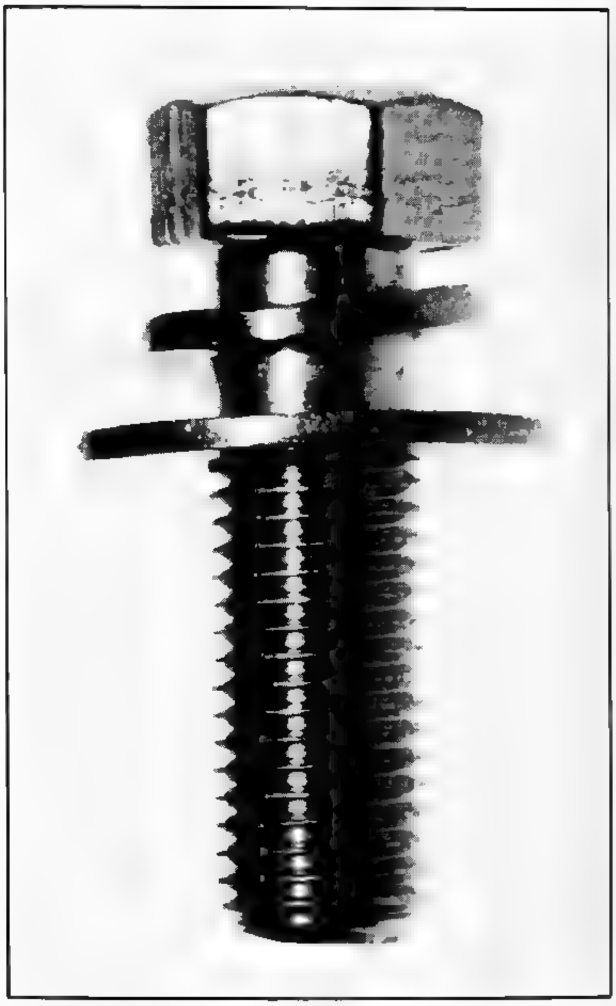
There should be a plain washer between every lock washer and the work face.



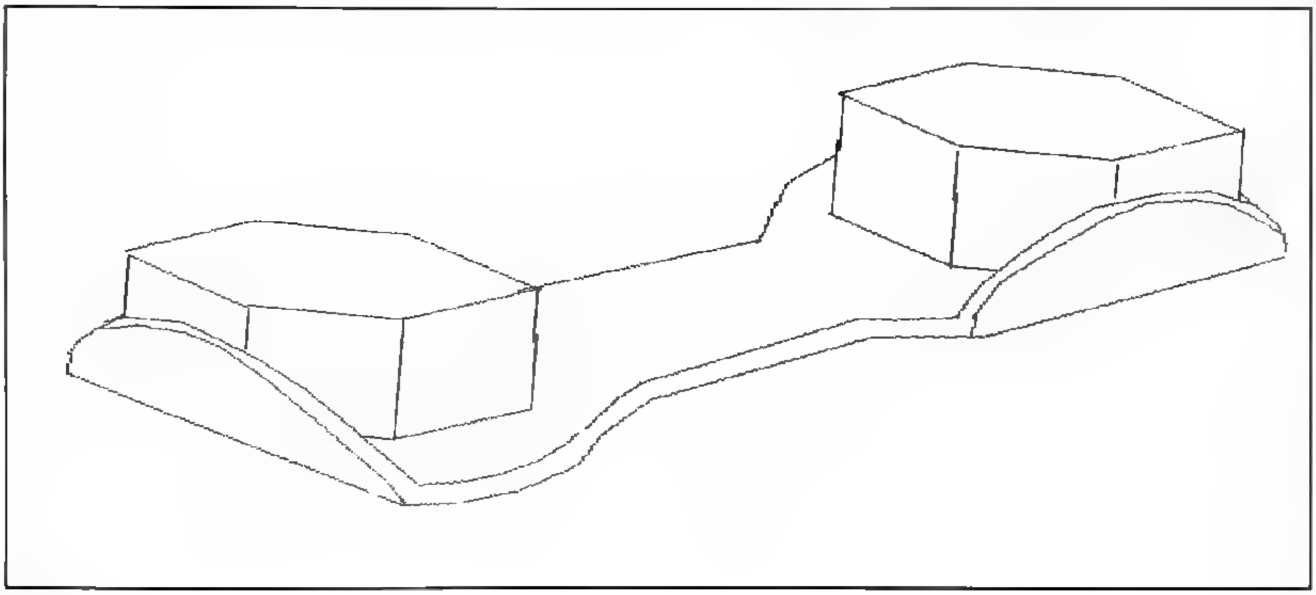
Basic types of lock washers.



Captive spring lock washer.



Captive spring lock washer with larger-area back-up washer.



The tab washer.

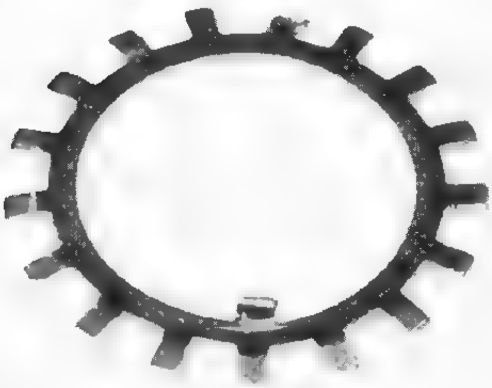
vent its physical departure. Even when the bolt breaks, a good job of safety wiring may save the day because the bolt head won't fall out and get jammed in the works.

Drilling the safety wire hole

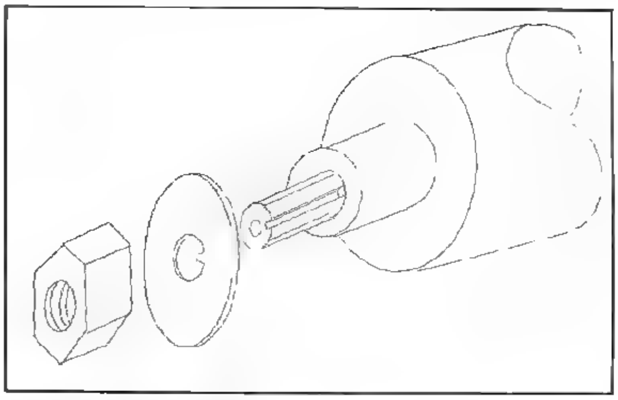
The use of safety wire is simple after you get the hang of it. The first thing to remember is that you need a hole through the bolt head. There are several ways of achieving this. The easiest is to buy drilled head AN and MS bolts. This is seldom practical, so you get to drill a lot of holes in bolt heads. Again there are several ways to do this. All of them work—sort of. Only these two methods work *well,* however:

Center punch the hole location in the center of one of the bolt head flats (we'll cover Allen bolts in a minute). Clamp the bolt in a drill press vise with the bolt head normal (perpendicular) to the drill bit in both planes and drill a #40 or 3/32 in. hole directly through the bolt head. Use a cobalt drill bit at relatively slow speed and some sort of cooling and lubricating oil. Withdraw the bit every so often to clear the chips.

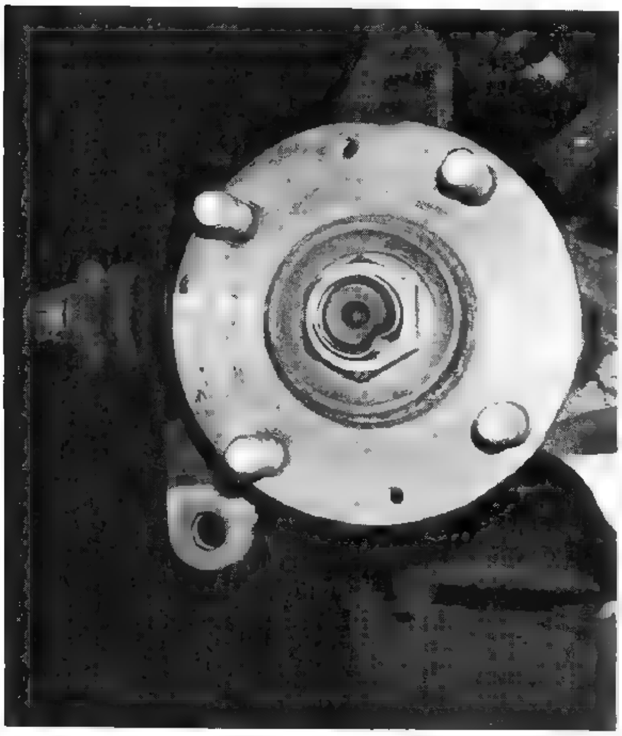
When you have finished drilling, deburr and chamfer both sides of the hole so that there will be no sharp edge to notch the wire for later breakage.



An alternative style of tab washer.



The keyed tab washer.



The key nut.

Safety wiring on a World War II airplane.

If circumstances require you to use a hand-held drill motor, use either a vari-speed or an air drill; keep the speed down and the feed constant. The primary sin committed against tiny drill bits is overheating them with excessive speed. Store your cobalt drills separately from your high-speed steel bits. They are expensive and you don't want to get them mixed up.

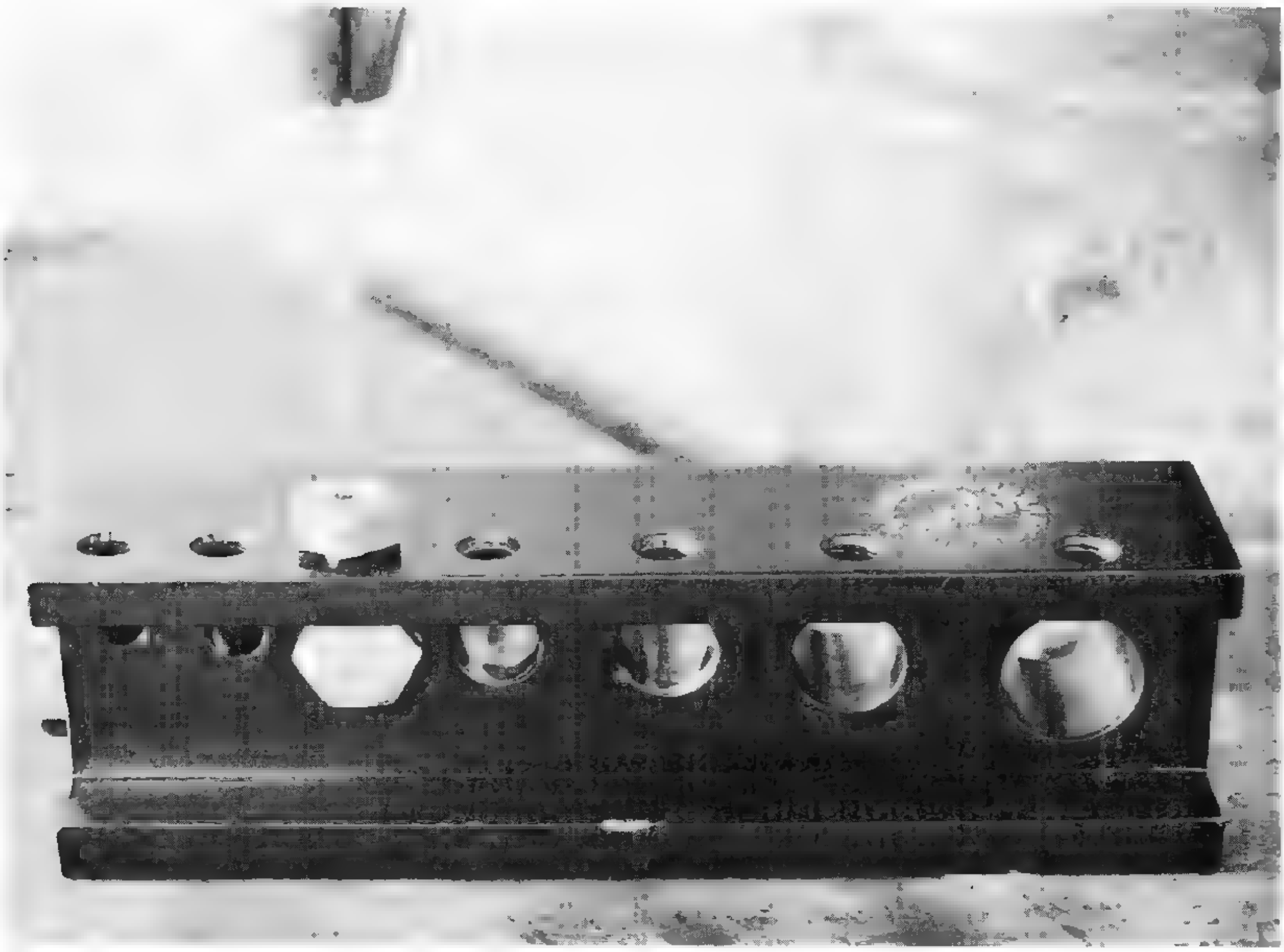
Every so often someone comes up with a gadget so simple and so brilliant it takes my breath away. Usually it also makes me feel really dumb for not having thought of it myself. I have been drilling safety wire holes in bolt heads for more years than most of you have been alive. During those years, I have broken countless drills (about twenty percent of them inside the damned hole), drilled countless off-center holes and wasted enough hours to write another book. I have tried several commercial drilling jigs, but not one of them has worked worth a damn. In fact, they have been so useless that I have always given up in disgust and gone back to the time-honored center punch, vise and drill press (or center punch, vise and hand-held drill motor) method of breaking drill bits.

Those days are over! A clever man named Tuck Jones now markets a drill jig that just plain works. You merely insert the bolt to be drilled into the appropriate hole, tighten the drill bushing down to hold the bolt still and start drilling. Of course, it is necessary to withdraw the drill bit every so often to clear the chips, it helps to use a lubricant and you do have to chamfer the holes when you get done, but the bottom line is good news indeed—the damned thing works! And it does Allen bolts. It even stores drill bits inside itself.

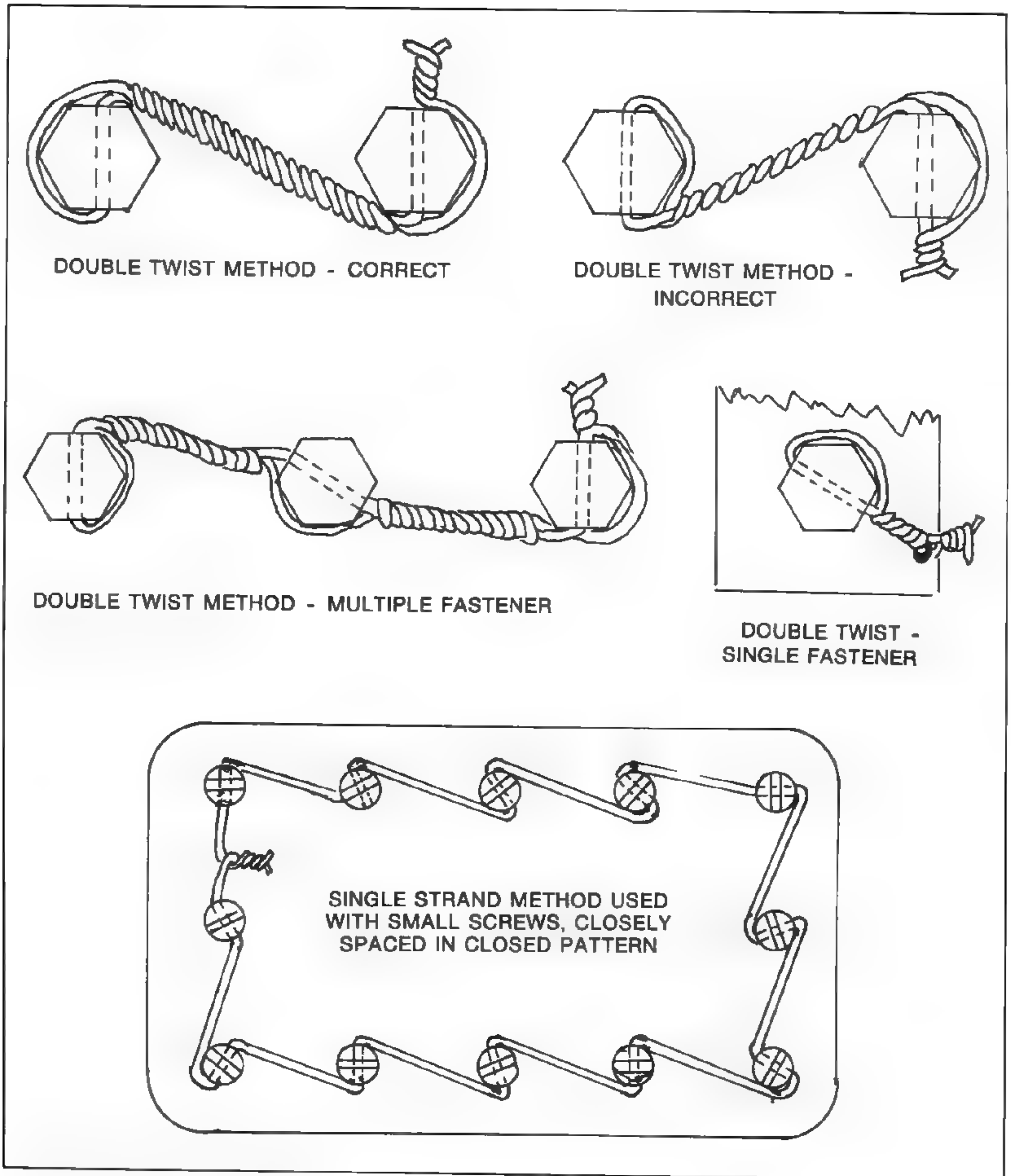
Currently, Tuck gets $50 for it. In my case it saved at least that in drill bits within a month. So thanks, Tuck Jones, and may profit reward your efforts (see appendices for address of Tuck Jones Engineering).

Allen bolts

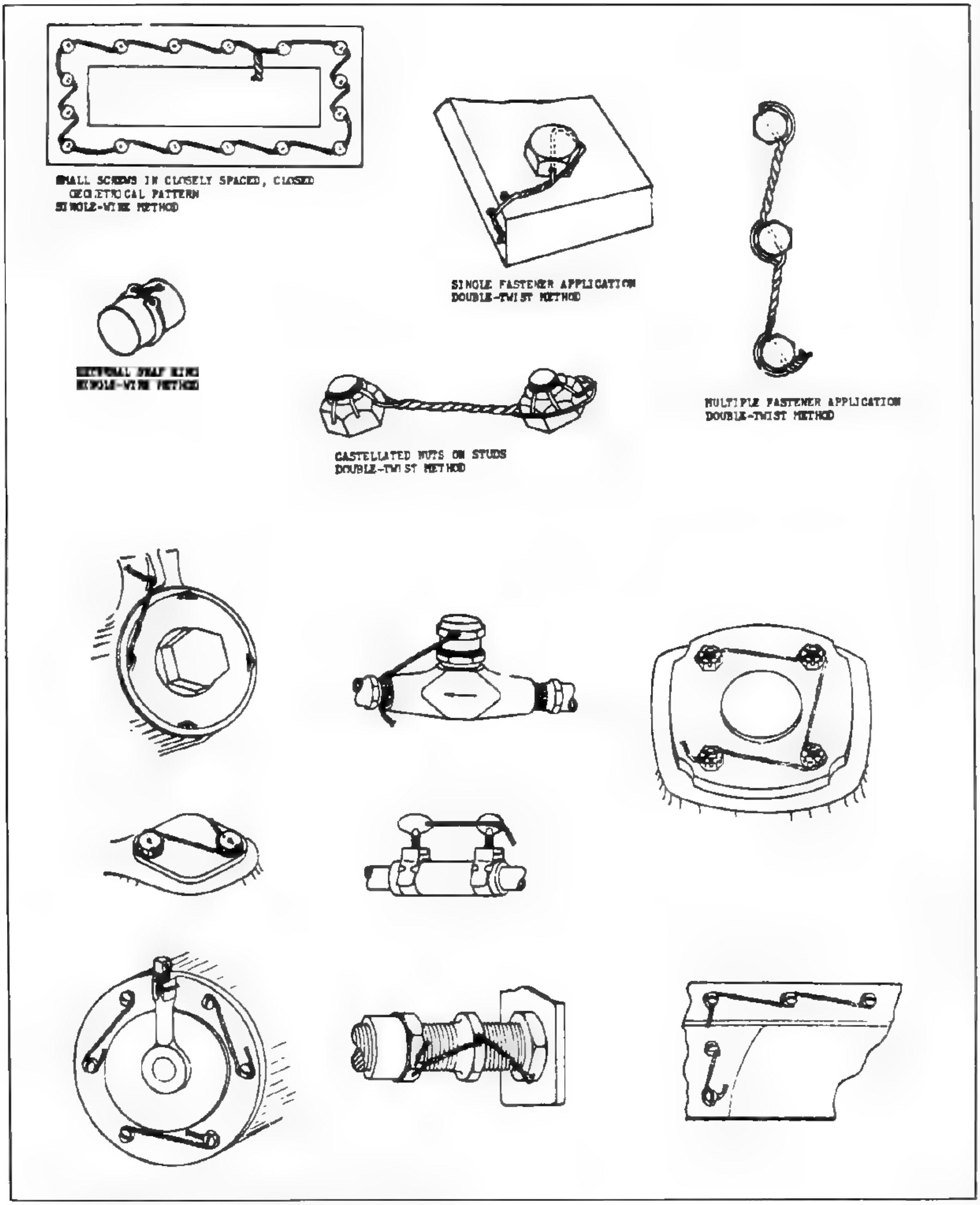
Allen bolts are difficult to drill for two reasons: they are case hardened and they are round. The roundness makes it difficult both to start the drill bit



The Tuck Jones Engineering jig for drilling bolt heads for safety wiring.



Textbook safety wire applications.



Standard applications of safety wire from the US Defense Department.

and to get the hole square with the world. The case hardening makes it difficult to center punch, and knocks the hell out of the drill bit. The solution to all of these problems is to grind a small flat on the bolt head parallel to one of the interior hex surfaces. This removes the case hardened layer and provides a flat working surface. There is nothing that can be done about the case hardened surfaces on the inside hex drive faces, except to orient the hole so that you drill through the inhex as vertically as possible.

Wire

Most of us who do a lot of safety wire use 302 stainless steel wire. It comes in the handy one-pound cans in diameters of 0.020, 0.031 and 0.041 in. The 0.031 in. diameter wire does virtually every job. At one time or another I have used almost any kind of wire that you can think of and it all worked. In Europe in the 1960s, when no one else used the stuff at all and there wasn't any to be had, I used copper wire from old generator windings and soft iron mechanics wire.

Twisting the wire

There are lots of ways to twist safety wire. The easiest is with a very expensive pair of safety wire pliers from Earl's Performance Products, or from your friendly aircraft tool store. If you work on race cars or airframes, they will pay for themselves in short order. For the occasional safety wiring job, a good pair of duckbill pliers does just fine. I have never been able to make the cheap, twisting devices work at all. The important things to remember are:

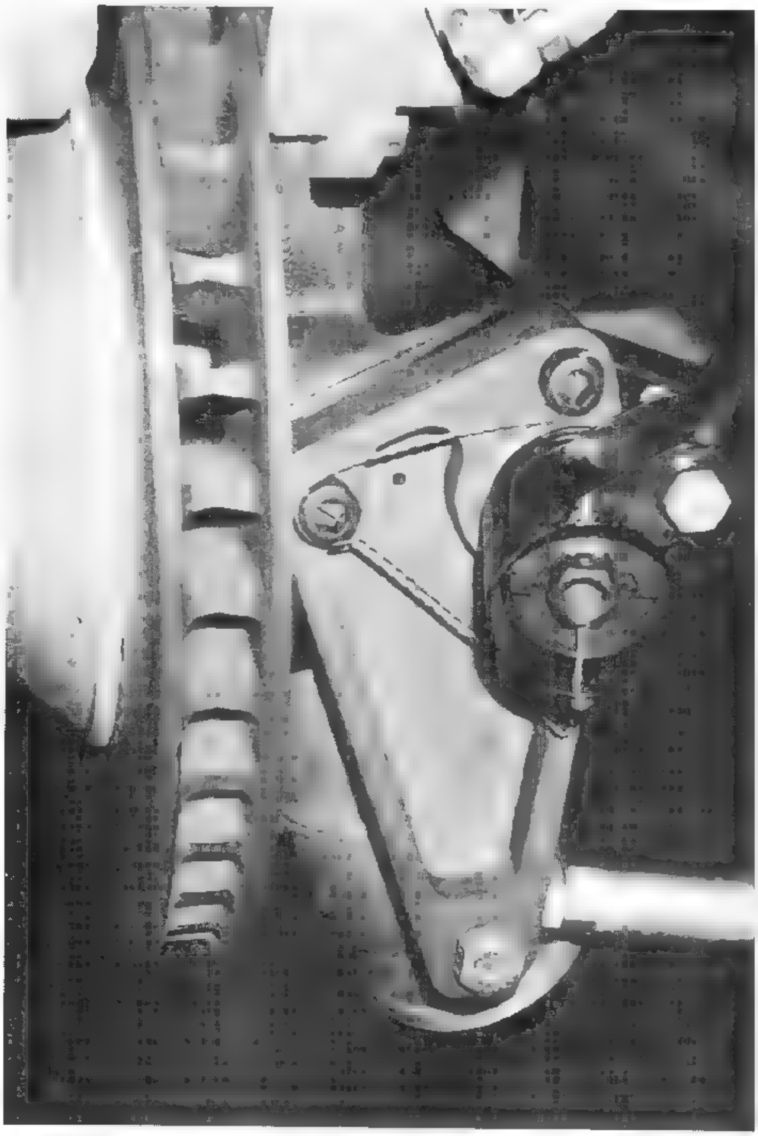
Cut the wire long enough to do the job—wire is cheap and nothing is more frustrating than to probe around in a difficult place until you are almost done and then find out that you don't have enough wire to finish the job.

Insert the wire through the hole in the bolt head and loop it around the bolt head (not over the top) in the direction that tightens the bolt. Pull it tight, estimate the length of twist you need, grab it with your pliers and start twisting.

Always install, pull and twist the wire in the direction that tightens the bolt.

The idea is not to twist the wire until it is tight but rather to pull the wire tight and twist it to take up the slack. Don't twist the wire too tightly and don't nick it with the pliers. If you do either, the wire will break—either now or later. I try to use a constant eight to ten turns to the inch. It is important not to use pliers with sharp gripping teeth as

Beginning to twist the safety wire.



Clean safety wiring work with a sculptured structure and bolting flanges on a 1967 McLaren.

they are guaranteed to nick the wire. A few minutes with a file or a belt sander will fix this situation.

I try to get the wire really tight. The best way that I have found to do this is to leave a little bit of untwisted wire—no more than ¼ in.—before the wire passes through the second bolt head. I then pull the wire tight (again passing the loose end around the bolt head in the direction of tightening) and twist the wire end while pulling, hard, to tighten the assembly.

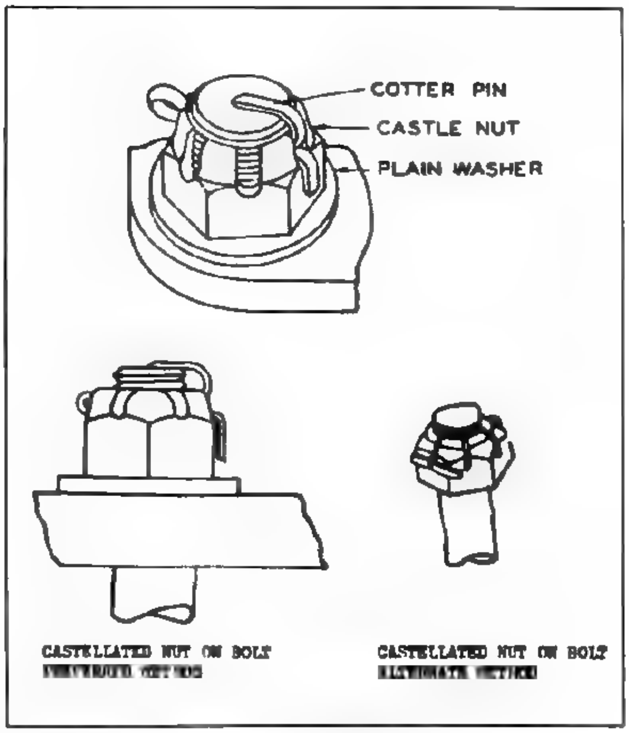
When you have finished twisting, snip the excess length off, leaving about ½ in. of twisted wire sticking out from the bolt head. Using pliers, bend this loop back on itself so that a rounded tip is presented to whatever part of your clothes or your body is bound to come in contact with the wire end. From sheer self-defense, this practice will eventually become second nature. It also makes the job look like someone cared.

When safety wiring, just as when pop riveting, it is good practice to carry a Dixie cup or coffee can around with you and to put the cut-off lengths of wire into it as soon as you cut them off. Otherwise they are bound to find their way into places where they do not belong and may cause damage. Again, after you have picked enough sharp pieces of wire out of your hands, your tires and so forth, this will also become second nature.

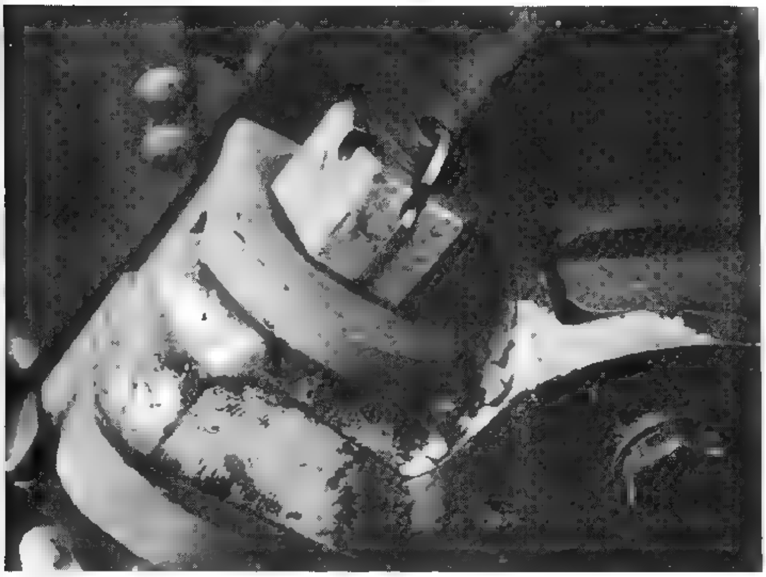
Don't even *think* about reusing safety wire. It just isn't that expensive.

Compromising

There will come a time when it seems impossible to safety wire a particular object. This is the time to throw the rule book out the window and compromise. If you cannot thread the wire through a bolt head because of an adjacent structure, loosen the bolt until you can and retighten it with the wire in place (before twisting). If the bolt is so buried that you cannot wrap the wire around the bolt head, bring it over the top—it beats no wire at all!



The cotter pin and the castellated nut.


The wrong way: the cotter pin is too small in diameter.
Roy Kiesling

Cotter pins

If a positive lock is required, and it often is, the much maligned and often overlooked cotter pin, used with a drilled shank bolt and a castellated nut, will provide it at a nominal cost. There are several things to remember:

Although AN and special-purpose bolts come with the shanks drilled for cotter pins, you still get to drill the threaded shanks of a lot of bolts. It is not easy to start a hole on a thread in such a way that the hole will be square with the world. To get the hole in the correct location and alignment, install the bolt and tighten the nut. Using an appropriate prick punch, mark the hole location. Place the bolt in a V-block or in a piece of old angle iron and solidly center punch the prick punch mark. Drill the bolt in the V-block or angle iron so that you have a reasonable chance of the finished hole being a diameter of the bolt and being perpendicular to the shank. If it is not, the pin will not line up with the nut castellations. Freehanding this job is almost guaranteed to end in disaster; it's a lot easier to do it right the first time. After you have drilled the cotter pin hole in the threaded shank of the bolt, deburr it.

Select the drill size so that the pin will be a reasonable fit in the hole. You don't want to have to

drive it in (or out), but you don't want it floating in space either.

Don't expect the castellations in the nut to line up with the hole in the bolt at the specified torque. They never do. Never loosen the nut to insert the pin. It won't hurt to tighten the nut enough beyond the specified torque so that they will line up.

Insert the pin and give the closed end a tap with a drift to seat it. Bend the ends over, one over the top of the bolt and the other axially against one flat of the nut. If you can wiggle the cotter pin with your fingers, you did it wrong. Take it out and start over. When you are satisfied, cut the excess length off. Try not to leave meat hooks, and put the cut-offs in your handy coffee can/trash bin.

You will often be required to insert cotter pins in awkward locations. The use of a longer than necessary pin will usually make this job a lot easier. If there is room to bend over the ends, there is room to snip off the excess.

Cotter pins are a one-use item. Don't even *think* about reusing them. If you do think about it, think about why you are using the pin to start with. Then think about what happens—both to the cotter pin and to the cut-off end—when the bent-over end that you want to straighten and rebend falls off. Finally, think about how much a cotter pin costs.

Other locking devices

Jam nuts

Probably the most positive of the conventional locking devices is the simple jam nut (or check nut). This is nothing more than another nut tightened down on top of the first. The first or holding nut is tightened to its predetermined torque and the second is tightened down on top of it. Tightening the two nuts against each other produces opposing and locked-in stresses in the bolt threads regardless of the residual stress in the bolt. The jam nut serves only as a locking device—all of the clamping load is carried by the first or holding nut. The jam nut is normally used to lock rod end bearings in place, but it works just as well in many other situations. For example, most professional racers use check nuts rather than cotter pins to retain both the pinion and constant motion shafts at the rear of their Hewland transaxles, simply because the check nut is faster when changing gears (and less likely to be forgotten).

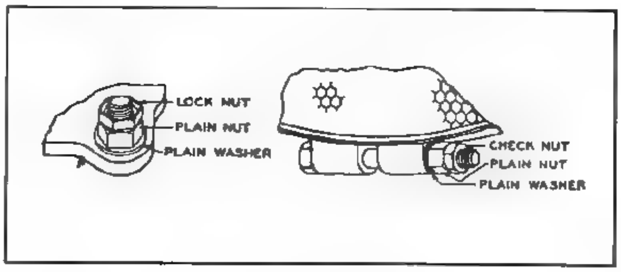
Shur-Lock system

The high-tech equivalent of the cotter pin is the Shur-Lock nut made by the Shur-Lock Corporation. I would use it in several applications—*if* I could afford them. I cannot so I do not.

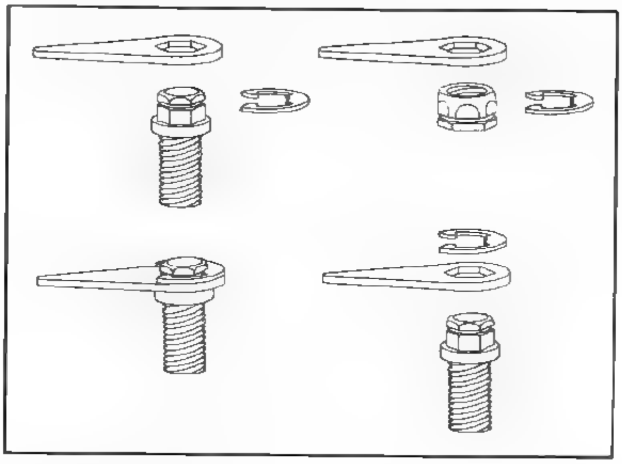
Commercial antirotation devices

Over the years I have seen any number of commercial antirotation devices. They all work by jamming some sort of bolt head extension against a nearby convenient solid object. None of them seem to have stayed on the market long. The one that currently seems promising to me is made by Stage 8 Fasteners of California. While it isn't suited to every application, where it will fit, it is both convenient and foolproof. It is particularly useful for exhaust manifolds, crankshaft dampners and the like. Although new to the market, some of the performance parts houses are stocking them.

Check nuts installed on Hewland transaxle pinion and layshafts in place of castellated nuts and cotter pins.



Typical use of check nuts as lock nuts.



The Stage 8 locking system.

Bruns lock

David Bruns is a very clever man. He is also, at times, emminently practical. Both of these statements are proven by the success of the road racing cars that he designs. The specific instance that I have in mind is the method that he uses to lock the female threaded wheel retaining bolt to the axle on his Swift Formula Ford, Formula Ford 2000 and Sports 2000 cars.

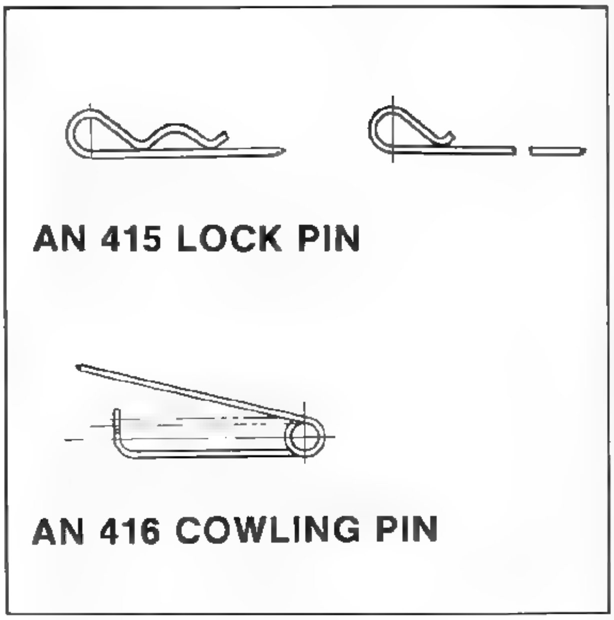
The actual axle is the male extension of the rear constant velocity joint. The axle fits through the hub bearing and splines through the drive hub. The wheel retaining bolt threads onto the axle, retaining the hub. It is treated with Loctite stud lock and torqued to 180 lb-ft. No one in their right mind is going to trust torque and Loctite in this application, and there is no practical way to fit a jam nut or to make the assembly self-locking.

What Bruns has done is clever indeed. He has drilled and tapped three #10–32 holes in the hub. The holes are located on a hole center diameter such that the head of a #10–32 socket-head machine screw will just barely clear the flats of the wheel retaining bolt hex head. They are spaced so that one of them will line up at the proper torque. This is an absolutely positive lock for a super-critical component and it is achieved at minimal cost. To my mind, that is what engineering is all about. As Nevil Shute once said, "An engineer is a man who can do for a dollar what any fool can do for ten."

Locking pins

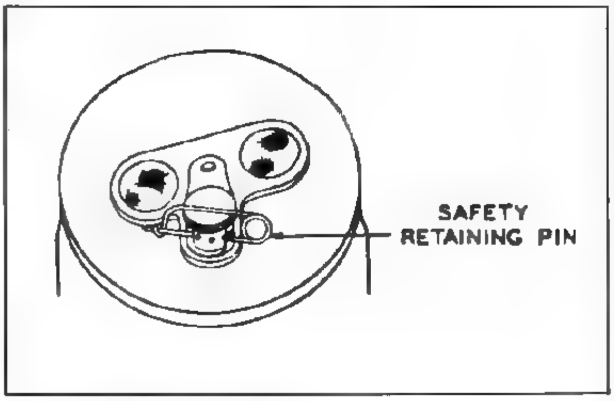
While I am on the subject of locks for threaded fasteners, the aircraft industry has spawned a pair of

The Bruns lock used on the real axle of the Swift family of racing cars.



AN–415 and AN–416 lock pins.

AN 415 used as safety on racing car wheel retaining nut.



Typical uses of AN lock pins.

convenient, clever and cheap locking (or at least antirotation) devices. These pins are inserted into a hole drilled in either the head or the end of the bolt shank. While they do not serve as positive locks in the same sense as a cotter pin does, they limit fastener rotation to a few degrees in a positive fashion. They are designated as the AN–415 lock pin and the AN–416 cowling pin.

The AN–415 lock pin is functionally identical to the lynch pin that we use in trailer hitches and so on, but smaller. The AN–415–1 version is designed to pass through two or more studs, while the AN–415–2 variety passes through one stud only. The AN–416 cowling pin is the aircraft version of the household safety pin. These little gems are not meant to prevent rotation of a bolt or stud but rather to retain a panel. We use them on racing cars to prevent the nut that retains the wheel from backing off all the way if it should loosen. While it is perfectly true that a properly tightened wheel nut cannot and will not loosen, we are all human and we do make mistakes. It is a great comfort to know that a quarter ounce of spring steel is living on the end of our axles to save our drivers from our mistakes.

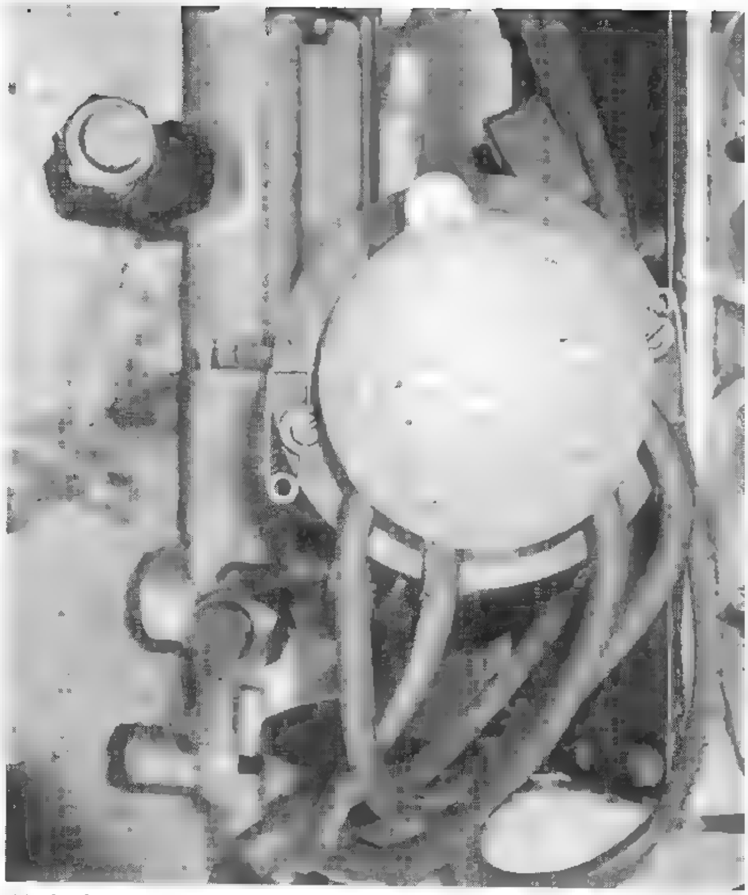
Also shown are the things we use in place of safety wire and cotter pins for all sorts of applications—basically because they are easily removed and reused. Some years ago it blew the kart racers away when my son Christopher showed up with nary a cotter pin on his sprint kart. Now the aircraft pins are pretty common in karting and are sold by all the kart shops at exorbitant prices—which may indicate a couple of things about karters and racers (it certainly indicates something about my business acumen).

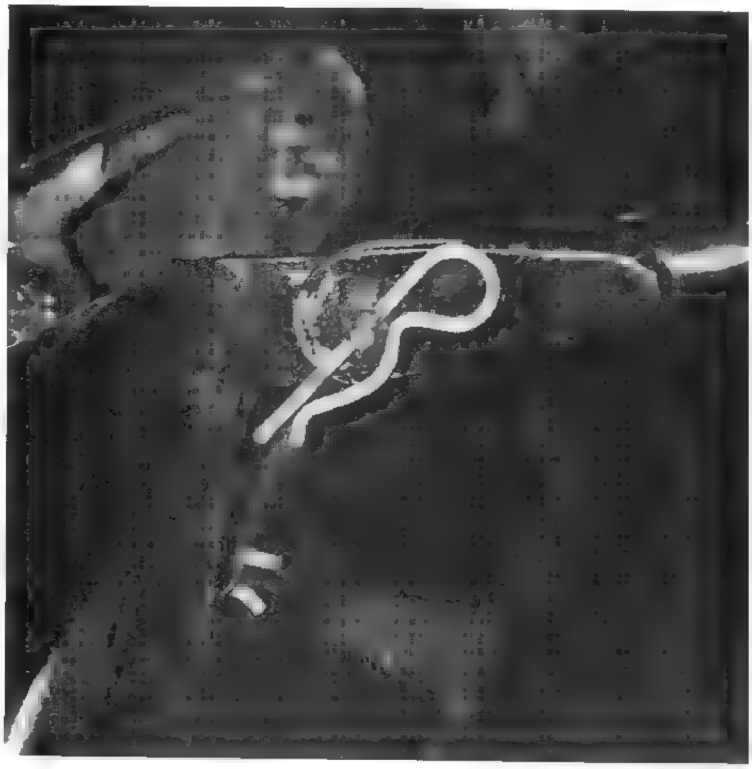
The Loctite story—
Or better racing through chemistry

The name of the Loctite Corporation has deservedly become synonymous with the term thread locking adhesive. In a 1970 magazine article, I called Loctite "the racer's salvation." It still is.

A lot has happened at Loctite since I first discovered their products in 1959. Loctite acquired the Permatex Corporation and integrated, streamlined and expanded both product lines. The quality remains superb. The products are now divided into five major categories: threadlocking, sealing, gasketing and retaining of cylindrical components, and bonding. Their catalog is complete, detailed, informative and free. Obtain it from your local distributor and read it.

In the beginning, we racers used only a couple of Loctite products—Nutlock (blue) and Studlock (red). No more! At the moment I carry with me four separate threadlockers, two sealants, one gasketing compound, two retaining compounds, a primer, two adhesives and an antiseize compound. I also carry Loctite's do-it-yourself O-ring kit, but seldom use it.


AN lock pin use.


AN lock pin use.

Essentially the Loctite compounds are anaerobic liquids. They fill the voids between mating parts and, in the absence of air and in the presence of metal they will cure or self-harden from the liquid state to a plastic film. The film adheres to the surface imperfections on both parts and must be sheared before the mating metal parts can move relative to each other. The mating parts are not only locked together, they are sealed against leakage and/or corrosion. Of the various compounds that are available I use the following:

Threadlocker #242 (blue). A general-purpose, medium-strength compound that, as the name suggests, does most jobs—so long as the thread in question is engaged for a minimum of 1½ diameters.

Threadlocker #271 (red). A high-strength compound for small fasteners. Loctite's definition of small is somewhat different from mine—to them any fastener under 1 in. in diameter is small. This compound is especially effective in aluminum and magnesium castings. After it has cured it is very resistant to fuels, oils and solvents. Where once I Helicoiled all of my Hewland transaxle and bellhousing studs, I now use #271. Contrary to popular belief, it has the same temperature range as #242 (to 300 degrees Fahrenheit). Its shearing or breakaway torque is, however, some 2½ times greater.

Threadlocker #262 (red). A hard-to-find, high-strength compound with "controlled torque tension"—whatever that may be. It is specifically compounded for high-strength fasteners that will be subjected to heavy shock/vibration loads and high levels of stress. I use it on ring gear bolts, universal and CV joint bolts, axle retaining nuts, flywheel bolts and the like. It has about fifteen percent greater breakaway torque than 271, and the same 300 degree Fahrenheit temperature limitation. Most engine builders use #262 to install all of the studs in the block. The general feeling is that red Loctite converts a Class 3 thread fit to a Class 5.

Threadlocker #290 (green). Believe it or not, this is a penetrating compound that locks *already assembled* threaded fasteners. I first ran into it under the name Loctite Super Wick-In in Australia. The fluid is drawn into assembled and torqued threads by capillary action and cures to a medium strength similar to that of #242. Like all of the Loctite compounds, #290 does not like oil or grease, so I spray the assembly to be treated with Loctite Primer N, Brake Clean or some similar aerosol cleaner before I apply the magic. Since I end up running a lot of cars that have been assembled by folks I do not know, I find Loctite #290 to be a great comfort.

Porosity Sealant #290 (green). Same stuff—we just didn't know it. In addition to locking threads, #290 will wick itself into any porous metal (including both ferrous and nonferrous castings and welds) and cures to form a tough, elastic and pressure-resistant seal that is resistant to fuels, lubes and solvents. It brushes on and the excess cleans off with a rag. I use it as a matter of course on the weld seams of all tanks, reservoirs and hard lines.

PST pipe sealant with Teflon. I no longer use Teflon tape. PST is a hell of a lot more convenient, easier to use, cannot shred and get into filters, jets and so on and seals better than the tape. I don't have to worry about which direction to wrap the tape either. PST is not just another pipe dope, it is anaerobic and completely fills the voids between the mating threads.

Retaining Compound #601 (green). This is serious stuff. It is designed to retain bearings, sleeves, pins, bushings or other cylindrical objects in bores that have become distorted, oversize or been otherwise buggered. It works! Shear strength is 3,000 psi and the temperature range is up to 300 degrees Fahrenheit. I use it where and when I need it. If things get really desperate, Retaining Compound #680 has a shear strength of 4,000 psi. Retaining Compound #620 has the same strength as #601 but is good to 450 degrees Fahrenheit.

My experience tells me (although the Loctite Corporation does not) that all of these compounds will also work on threads. It should be noted that disassembly will not be easy and that the shear figures that I have given are for steel. Used with either aluminum or magnesium, the figure drops to 600 psi in all cases. When the designer has been cheap and used a right-hand thread where he should have used a left-hand thread, I find that #680 works wonders until I can figure out a mechanical lock.

Gasket Eliminator #515. I used to carry around a whole bunch of gasketing glops, ranging from the various silicone sealants to Hylomar. No more! Now I carry Gasket Eliminator #515 with a tube of their (or Permatex's) high-temperature silicone sealer as a back-up. The #515 has two major advantages over anything else that I know of: it peels off the disassembled surface as a ribbon rather than having to be scraped or scrubbed off *and* it fills voids between mating surfaces better than any other gasket goop. Used with Loctite Primer N, it will fill the seal voids up to 0.050 in., which makes it a good damaged surface repairer as well as a gasket eliminator.

Nickel anti-seize #771. When I cannot find my very favorite antiseize compound (Copaslip, which is hard to find, and expensive but worth it) I use Loctite #771 which is probably every bit as good.

Primers. Loctite primers decrease cure time, clean parts and improve the void-filling capability of the various anaerobic compounds. They are a minor pain to use, though. I carry Primer N with me and use it when I think I need it—like when I want the Loctite to cure in a hurry.

Speedbonder #319. This is a general-purpose bonding agent with good impact resistance and a

very fast cure. I use it to rig quick and dirty holding fixtures. Used with Primer N, bonded metal parts can be handled in less than one minute and reach fixture strength in fifteen. It also fixes broken china and other items around the house, thus earning valuable points with your housemate.

Quickset adhesive #404. I use it to splice my own O-rings from the Loctite O-ring kit. I used to carry a Parker O-ring kit around. I always had every O-ring except the one that I needed. Now I carry the specific O-rings that I think I might need (in sealed containers) and the Loctite do-it-yourself kit. It will not make dynamic seals, but it works just fine for static O-rings.

Quickmetal Press Fit Repair. This is a new product that I have been carrying for about a year. It is supposed to be a *permanent* repair for worn shafts, housings, keyways and the like. I hoped that I would never need it, but I had a hunch that one day it was going to save my butt.

Sure enough, it did. In the autumn of 1986 we lost the female thread of a critical aluminum casting insert in a crash. We had lots of inserts (it was the casting itself that was damaged), but there was simply no thread left to accept the insert. There was also no TIG welder, no lathe and no mill available. We tapped the casting oversize, coated the new threads with Press Fit Repair, inserted a similarly coated insert and waited for the Loctite to set. When it had, we installed the bolt, drilled down through the casting and the bolt, tapped the new hole in the casting and threaded in a #10–32 bolt to lock the whole mess. We were in the hinterlands when this happened and couldn't get to a real facility for four races. The temporary fix worked just fine and was sound as the Swiss franc when, weeks later, we took it apart and did it right. I told one of the Loctite field engineers about this one and he just shook his head and walked away.

Instant Gas Tank Repair. This is a two-part ribbon epoxy that fixes leaking metal gas tanks without draining them. I have a couple of these kits in the glovebox of the truck—they also work on metal fuel churns (*nobody* enjoys welding on a fuel container, even when it is filled with water).

Killing the Loctite bond

Every so often we run into a situation where cured Loctite prevents us from removing something—usually a cheap and nasty internal wrenching bolt that has stripped its in-hex. The solution, since all of the Loctite compounds are temperature limited, is heat—lots of heat. Steve Jennings taught me that if the head of the offending cap screw is heated to bright red with a small-tipped welding torch (to localize the heat), allowed to cool and struck a sharp blow with a hammer, the damned thing will usually come out.

Loctite and slippery coatings

The Loctite compounds are designed to work on clean metal; they will not function on either oily or dirty metal. This means that *both* male and female thread surfaces must be cleaned prior to application. Since most of our solvents (and the racer's normal cleaning agent, gasoline) leave an oily film on everything that they touch, it is easy to achieve a false sense of security with Loctite. I tend to use one of the brake cleaning compounds to clean the parts, for the simple reason that they work and are both easy to find and to carry around. Use whatever you prefer, but get all of the oil, grease and dirt (and all of the old Loctite) off both the male and the female parts—and dry the parts—before you apply Loctite.

How much?

Most people seem to use Loctite in accordance with the "If a little bit is good, a lot is better" theory. It doesn't work that way with Loctite. All that is required is a single drop. More will merely squeeze out as the bolt is tightened and get between the pieces to be clamped or between the bolt head and the work face. In either case you will end up with an unplanned layer of hardened adhesive where you don't want it. The solution is simple—read the directions and use only the recommended one drop (or a thin smear for Retaining Compound #601).

How long?

As near as I can tell, so long as the operating temperature stays somewhere near their prescribed limit, all of the Loctite bonds last indefinitely. The shelf life is also indefinite. Unlike virtually every other liquid, when storing Loctite, we are looking for the maximum exposure to air. Storing the bottles on their side will enhance the shelf life, as long as the caps are on tight. When the dispensing hole in the squeeze bottle gets clogged with cured Loctite (and it will), remove the bottle top and run a needle or a piece of safety wire through the hole. I buy my Loctite in either the 10 cc or the 24 cc sizes simply because I know that if I buy the large economy size it will sprout wings and fly away long before I have used it up.

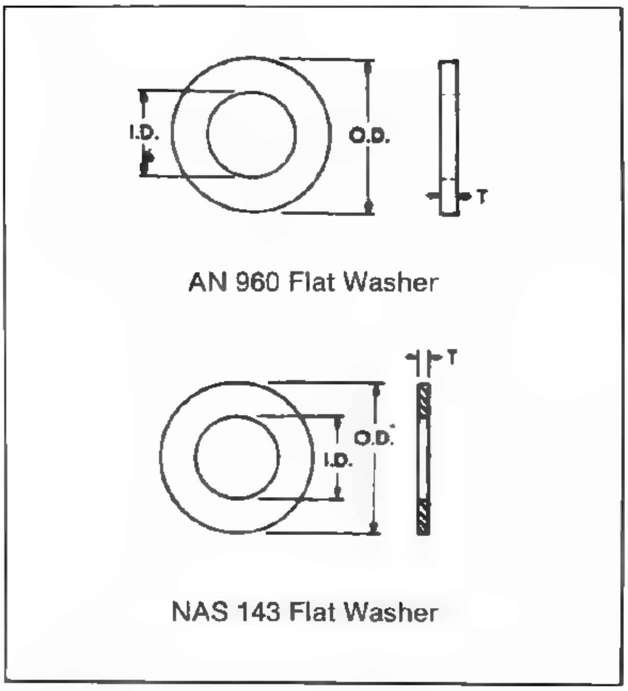
8

Washers and miscellaneous fasteners

Flat washers

Flat washers serve as bearing surfaces to prevent bolt heads and nuts from digging into the work surfaces. They also allow more accurate installation torque or strain measurements. Finally, they serve as shims. The only functional requirement is that they have a hole the right size, that they be flat—which is just about guaranteed by the manufacturing process—and that they be hard enough to prevent the nut or bolt from sinking into them under load.

For all applications where the bolt is loaded in shear and for most tension applications, I prefer to use the AN/MS items. The part numbers are AN-960 for the full thickness (0.063 in.) and AN-960L for the thin (0.031 in.) variety. They are also available in large outside diameter configuration (AN-970) for use with wood or fiberglass. These washers are lighter and a damned sight more handsome than SAE washers, and are manufactured in both steel and aluminum. I used to use the aluminum washers in shear applications, but I have decided that the weight saving is not worth the logistic trouble and the possibility of someone installing one in the wrong place. In severe tension applications, I use hardened and ground machine washers from the machinery supply house, or NAS washers.



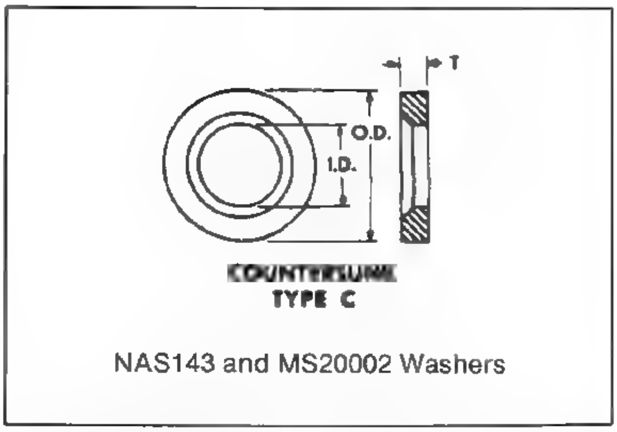
AN, MS and NAS flat washers. Top, AN-960 non-heat-treated washer. Bottom, NAS-143 heat-treated washer.

Countersunk washers

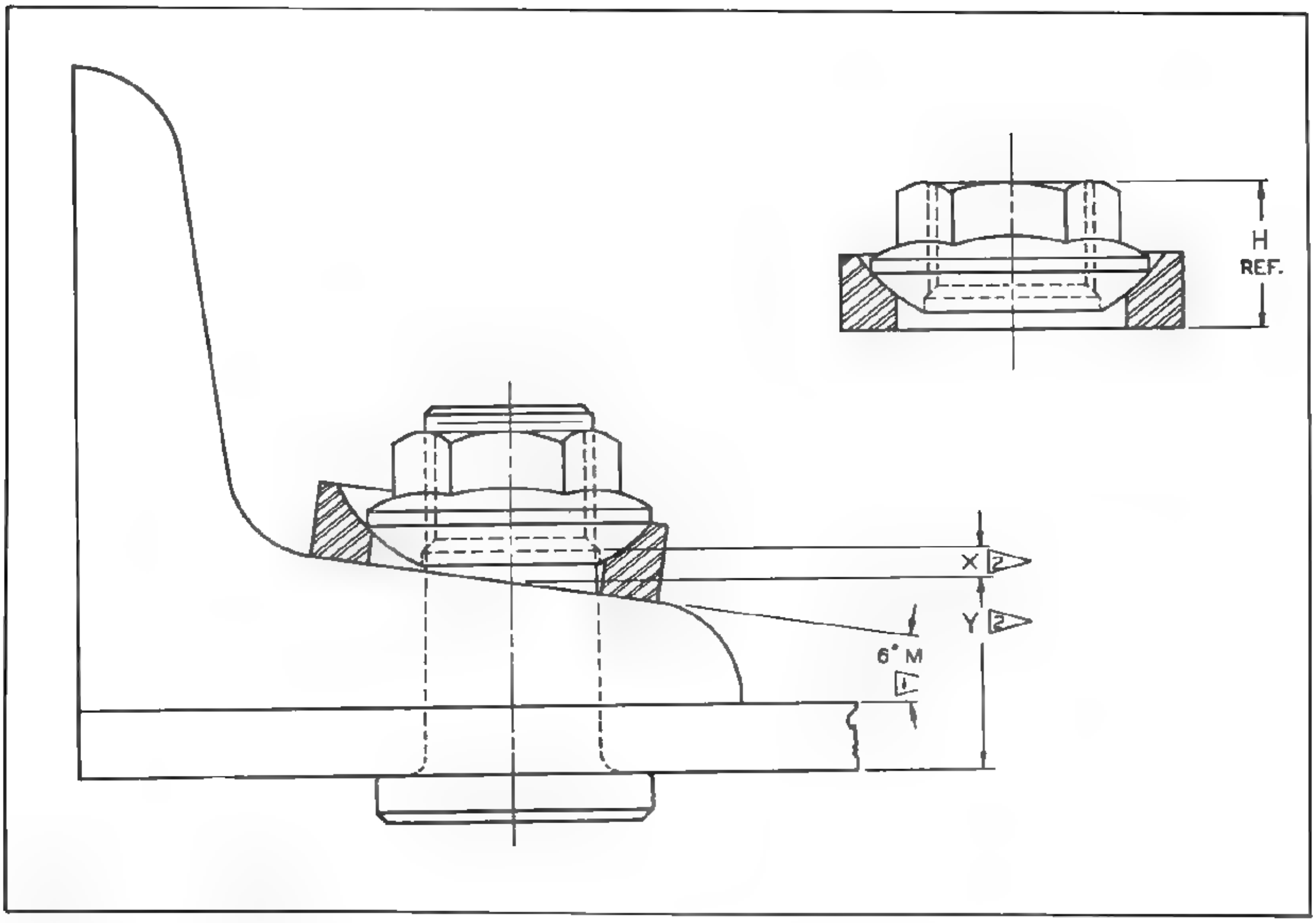
With high-strength NAS bolts, the large radius between the head of the bolt and its shank makes it necessary to use a countersunk or beveled washer. The part numbers are NAS-143 and MS20002. These washers are heat treated to prevent the bearing surface of the bolt from digging in under the increased preload of these bolts. They are available from AN/NAS bolt suppliers.

Ball and socket washers

We all know that we are supposed to drill all of our holes absolutely normal to the surface of the parts to be clamped—and we almost always manage to reach at least a close approximation of this goal. Every so often, however, we manage to come up with an installation where the work surface is not perpendicular to the bolt axis, and we cannot spot face the hole. If there is any misalignment at all, it is important to realize it and to make allow-



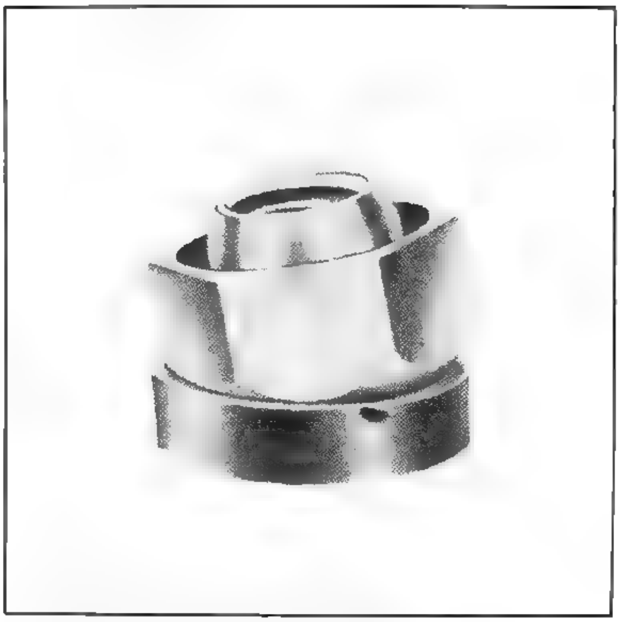
NAS-143 and MS20002 countersunk washer for use with NAS high-strength bolts with large head-to-shank radii.



SPS ball and socket washers.

ance for it. Otherwise, tightening the bolt will place the bolt in bending, which will produce a tension stress that was not allowed for in the design. This unintentional stress is cumulative with the installed stress but it will not show up on the torque wrench. Neither will it appear on either the strain-measuring micrometer or the load-sensing washer. Early fatigue failure is the only possible result.

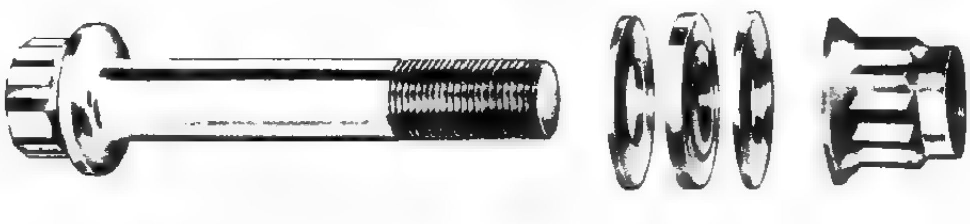
Admittedly, we should never end up in this sort of situation, but from time to time we just do. It is nice to realize that the high-priced help in aerospace experiences the same problems. In the old days it happened as a result of tapered wooden spars and longerons. Now it happens with tapered forgings and castings. The difference between us and the aerospace people is that they have the horsepower to make sure that the fastener industry comes up with an engineer saver when they need one. That is what I call a useful technology spin-off! The low-cost fix is the AN-950/AN-955 ball and socket washer. The high-dollar solution is the nut and captive washer ball and socket setup.



Ball and socket captive washer nut.

Preload indicating washers

There are many bolt applications in aerospace where it is necessary to accurately measure the



WASHERS — PRELOAD INDICATING			
		PLI 22, 26, 30	Preload ndicating Washer Assemblies, 220, 260, and 300 KSI Levels
		PLI L22	Preload Indicating Washer Assembly, for use with SPS LWB 22 Fastening System, 220 KSI
		PLI	Preload Indicating Washer Assemblies, 80, 100, 125, 160, and 180 KSI Levels

Preload indicating washers.

The Belleville conical spring lock washer.

installed tensile stress or preload of a bolt, but where it is not possible to measure the bolt strain. The preload indicating washer was developed for these applications.

The system consists of a series of three washers and a stress-sensitive ring. As the bolt is strained, the stress-sensitive ring collapses. When a predetermined level of internal tensile stress is reached in the shank of the bolt the ring collapses, the washers bottom and the assembly changes color. Each assembly will indicate only one predetermined level of stress, and the washers are not reuseable. The washers are color coded according to the stress level that they will indicate. They are available at vast cost from your SPS distributor.

Lock washers

We have already discussed the family of lock washers in chapter seven.

Belleville washers

If I should ever think that I need a lock washer, I will use a Belleville-type washer. These full-cone washers are made of thick spring steel and, when compressed, develop enough compressive stress to do some real good. They are available from good fastener houses and are sort of a poorman's place bolt. In fact, several manufacturers use them for special applications, like holding crankshaft torsional vibration dampners on.

Pins

In many applications we either do not have room for a conventional fastener or we do not need one. When the load is in shear and the requirement is to hold parts together, to locate them or to pro-

vide a fulcrum—and when no clamping force is required—we we can often save ourselves both time and money by the use of one of the ever-expanding family of pins, either conventional or quick-release.

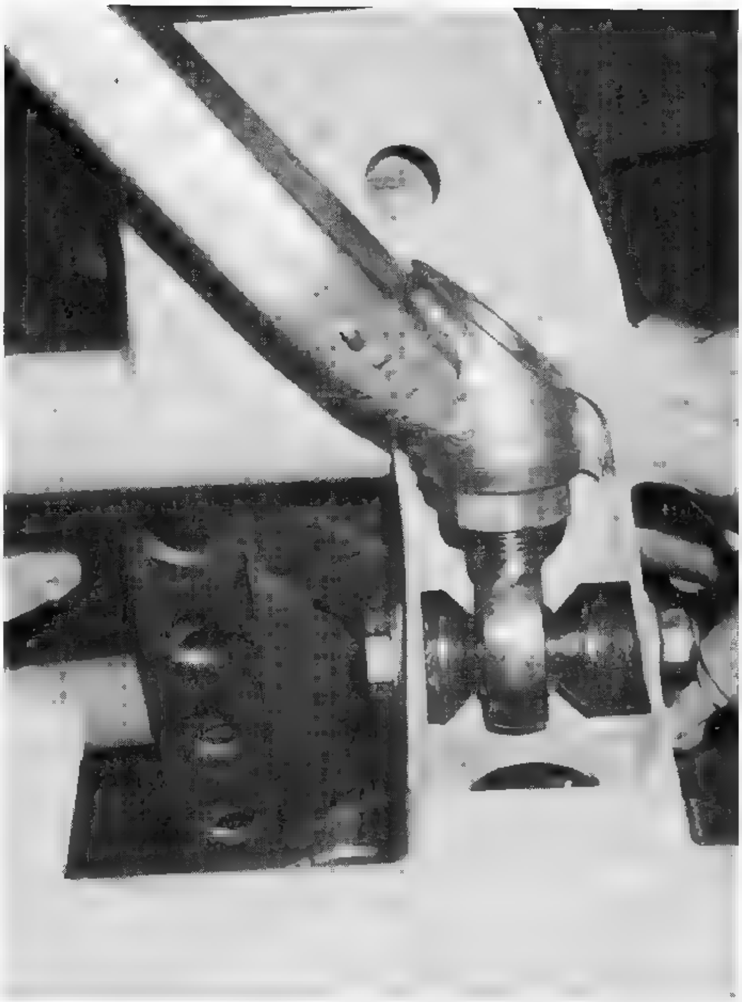
Clevis pin

The most simple pin that I use is the clevis pin. The clevis pin is an unthreaded pin with an upset head at one end and a chamfer at the other. The chamfered end is provided with either a cotter pin hole or a retaining ring groove. In use, the clevis pin is inserted into a prepared hole and retained by a washer and either a cotter pin or a retaining ring. Clevis pins are widely used in both the automotive and general aviation industries as retaining devices and as light- or intermittent-duty fulcrums. For many years we used them as the fulcrum for the clutch release arm on Hewland transaxles. Their major advantages are their low cost, the ease of assembly/disassembly and the simple fact that, unlike a bolt and even the most expensive self-locking nut, they cannot loosen.

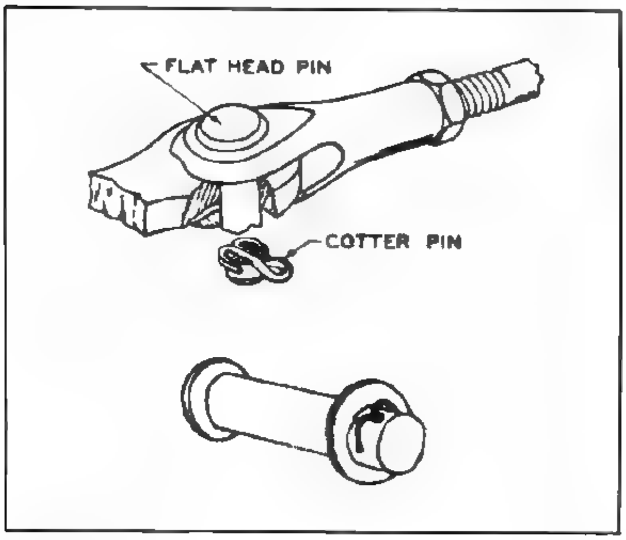
Taper pin

The standard taper pin, manufactured with the Morse taper of ¼ in. per foot, has been used from the very beginning of the machine age to positively retain pullies, wheels and levers on shafts and to fasten shafts together. It offers a precise fit and easy disassembly.

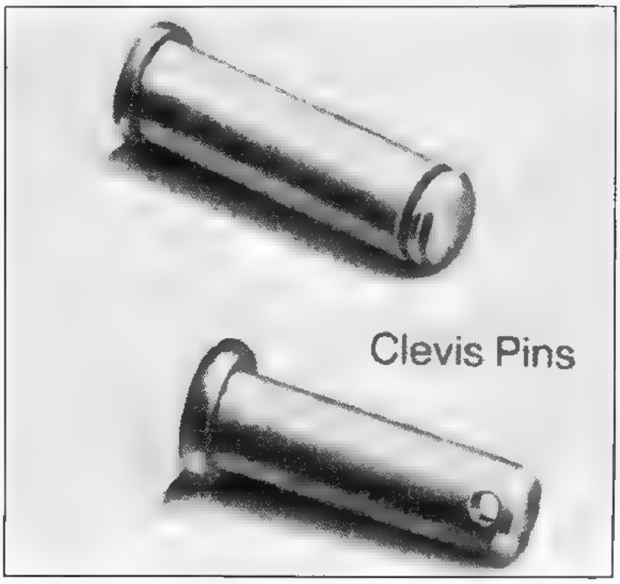
I do not use standard taper pins because I am afraid that under some combinations of load and vibration they might loosen and fall out. (You will have noticed by now that I tend to worry a lot.) The AN-385 pin is identical except that it features a safety wire hole at the large end so that the pin cannot fall out. It can still loosen, so I don't use it either. I do use the AN-386 variation, which features a threaded section at the small end so that it can be positively retained with a self-locking nut. I



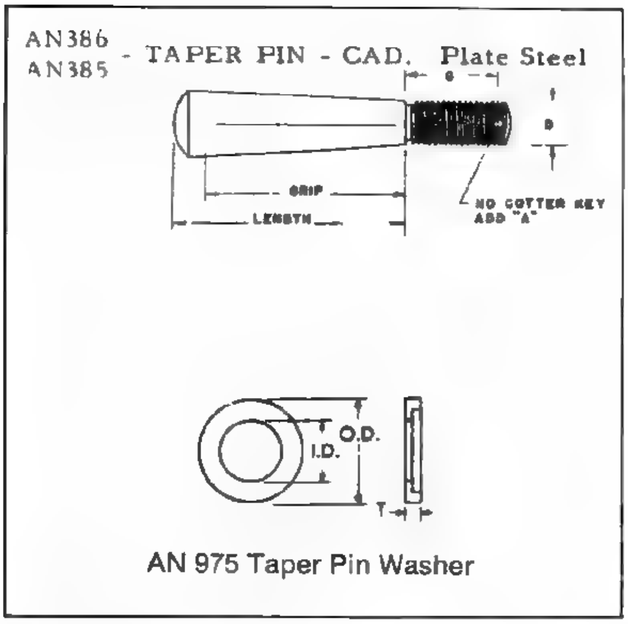
Tapered washers used to convert a standard rod end bearing to semi-high angularity.



The clevis pin.



evis pins.

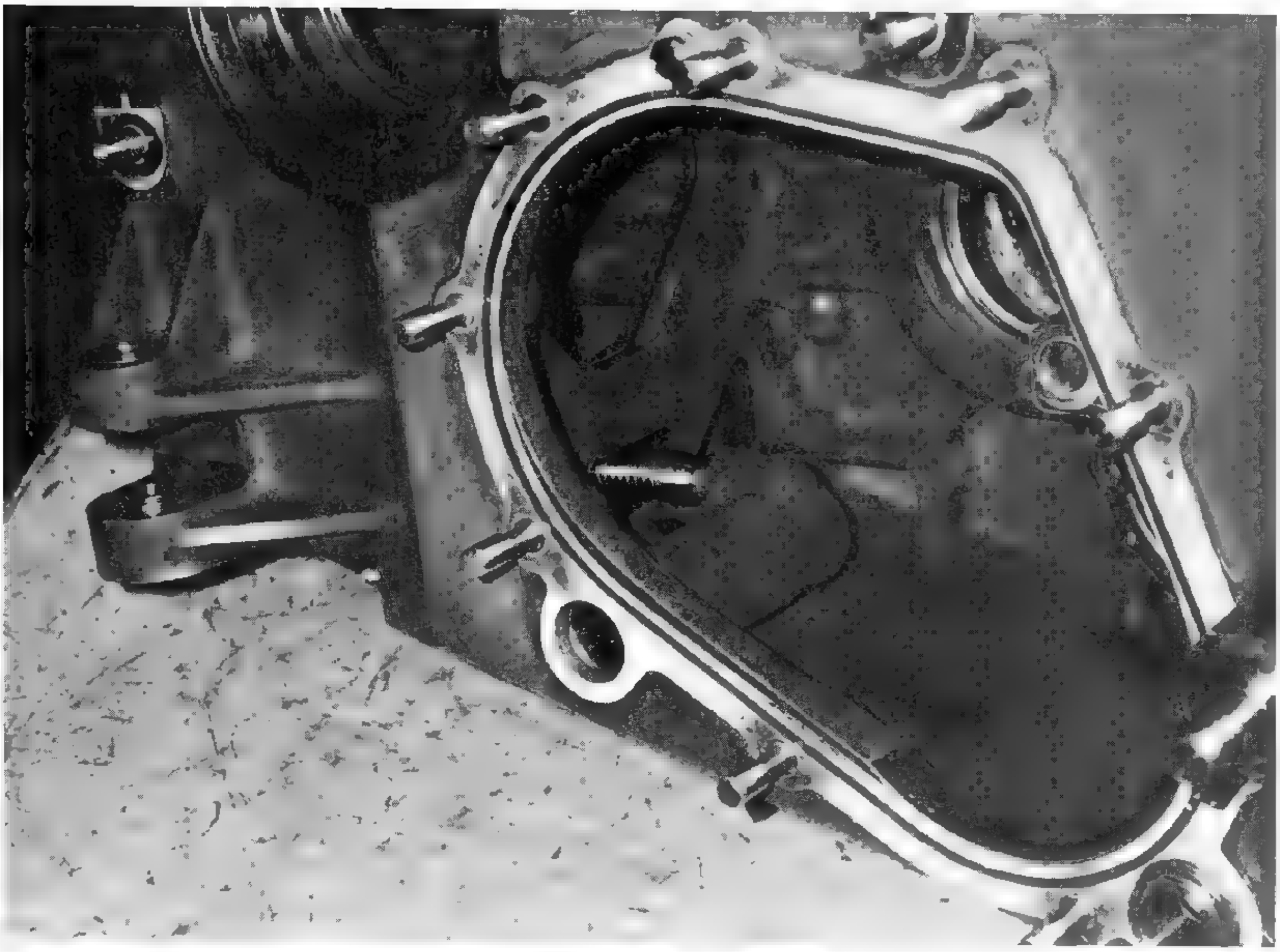


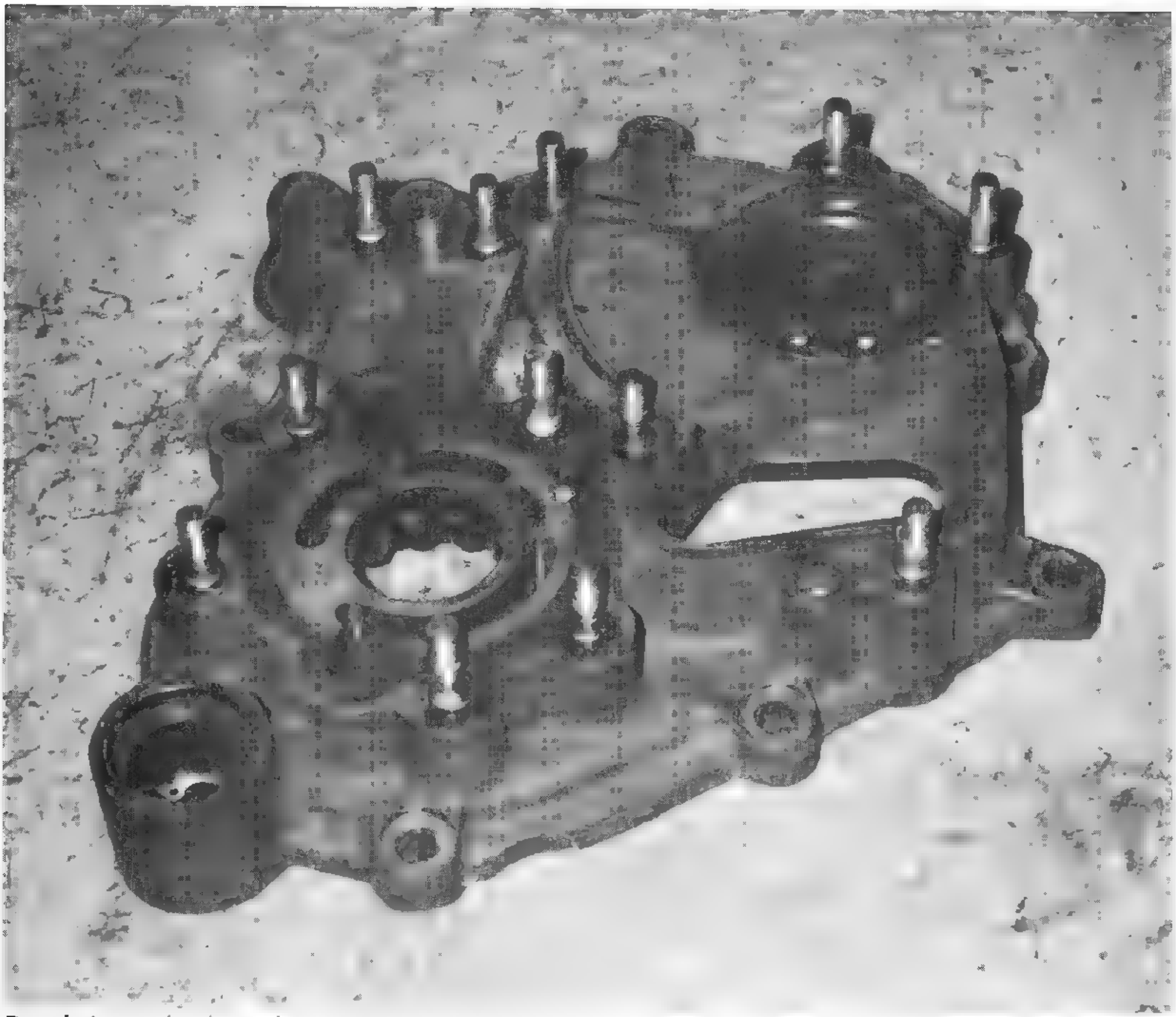
AN taper pins.

use these to retain nesting cylindrical parts when I want a tight assembly with zero radial and longitudinal play—like shift linkages. Unfortunately, the AN–386 pin uses the Browne and Sharpe taper of ½ in. per foot, so you cannot use a standard Morse taper reamer.

Dowel pin

I have gone to some length to point out that the intended function of the threaded fastener is to clamp parts, not to locate parts with respect to each other. Well, the intended function of the dowel pin is to locate, not to clamp. Available in both hardened and unhardened steel, the dowel pin is permanently pressed into a reamed hole in one of the members to be located (I use Loctite #601 to make sure). The protruding part of the pin, which has a chamfered end, fits into a matching but slightly oversized hole in the mating part. Needless to say, adequate location by means of dowel pins calls for the use of several pins and some pretty precise machining. You are not going to do any good at all with a hand-held drill motor, and it is pretty


Use of dowel pins to locate side cover on Weismann Formula One transaxle.

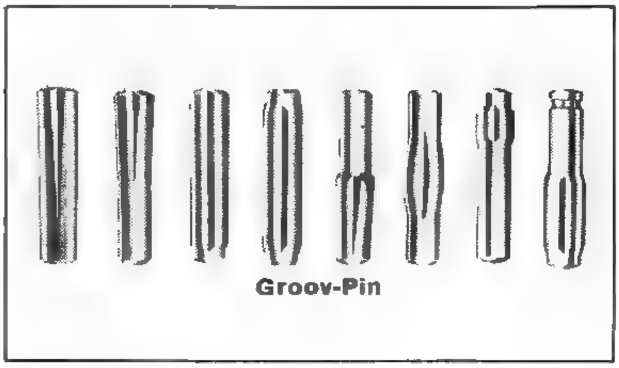


Dowel pins used to locate housing to Weismann gearcase.

unlikely that you will succeed with a drill press. I clamp the parts in their desired location in a mill, drill through undersize and then ream both holes to final size. I use dowel pins, for example, to locate the shift finger housing and rear cover of Hewland transaxles to the gearcase so that the down load from the wing doesn't cause fretting and oil leaks.

Solid straight pin

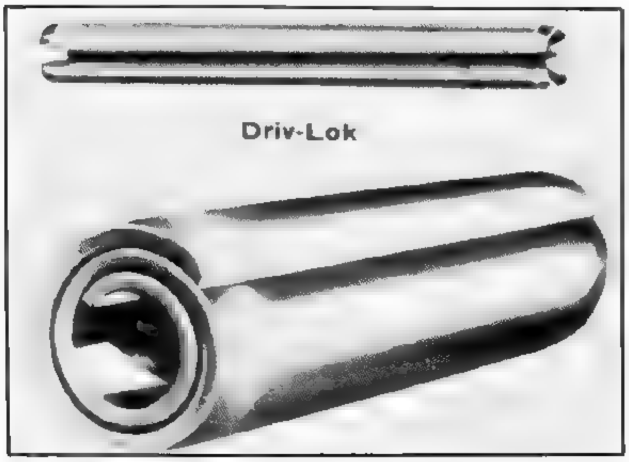
The solid straight pin is widely used in production as a low-cost, quickly assembled shear fastener. It is meant to be a tight fit in drilled holes and is usually retained by peening over the end. They are also available with a nylon insert retaining device. To my way of thinking they are nothing more than a crude rivet. I don't use them—simply because I don't do anything that would tolerate a fastener this imprecise.



Types of grooved pins.

Grooved straight pin

The grooved straight pin can be a useful device. These are cylindrical straight pins onto whose surface carefully designed grooves have been impressed. When these pins are driven into holes of the proper diameter, the elastic action of the grooved section under compression locks the pin firmly in place. The pins have been standardized in eight different groove configurations. Widely used in production as pivot pins, shear fasteners, dowels and stop pins, they are available in a wide range of materials and in diameters from 1/64 to ½ in. and in lengths from ⅛ to 4½ in.



The Driv-Lok, top, and Spiral-Lok, bottom.

Spring pin

The spring pin, usually called "roll pin," is a tubular device formed from either coiled or slotted spring steel. Spring pins are designed to be driven into holes with diameters that are slightly smaller than the pins. Spring pressure exerted by the expansion of the pin against the wall of the holes holds them firmly in place. They are easily inserted and removed and are almost infinitely reuseable.

I like spring pins. They are typically stronger in shear than the equivalent solid straight or grooved pin, and are less critical of hole size than other pins. They can also be nested to provide still greater shear strength, can be safetied in position with simple wire and can be shortened (by grinding) to suit individual requirements. I carry an assortment of long ones around with me as an emergency repair item and, while I don't need them very often, when I do they are a genuine lifesaver.

Quick-release pins

Most of the family of quick-release pins is shown here. The simplest of them is the "push-pull" pin. It is a simple straight pin with a ring mounted on its head, and is designed to be used as a quick-release holder on the ends of rods or shafts. The pin is inserted through a hole in the end of the shaft and locked in place by snapping the ring over the end of the shaft and against the pin. This pin is functionally a quick-release cotter pin and is often used as hold-downs for hoods, engine covers and the like. I never liked them until Precision Prepara-

SPRING PINS	MS 9047 and MS 9048	Spring Pins, Alloy Steel AMS 5120 or AMS 5121
SPRING PINS	MS 16562	Spring Pins, Carbon Steel or CRES Steel
SPRING PINS	MS 171401 thru MS 171900	Spring Pins, CRES Steel AMS 5506
SPRING PINS	NAS 561	Spring Pins, Carbon Steel or CRES Steel
DOWEL PINS	MS 16555	Dowel Pins 0.0002" Over Nominal Size Carbon Steel, A loy Steel, CRES Steel
DOWEL PINS	MS 16556	Dowel Pins, 0.001" Over Nominal Size Carbon Stee , Alloy Steel, CRES Steel
DOWEL PINS	NAS 607	Dowe Pins Case-Hardened Alloy Steel

Spring pins and dowel pins.

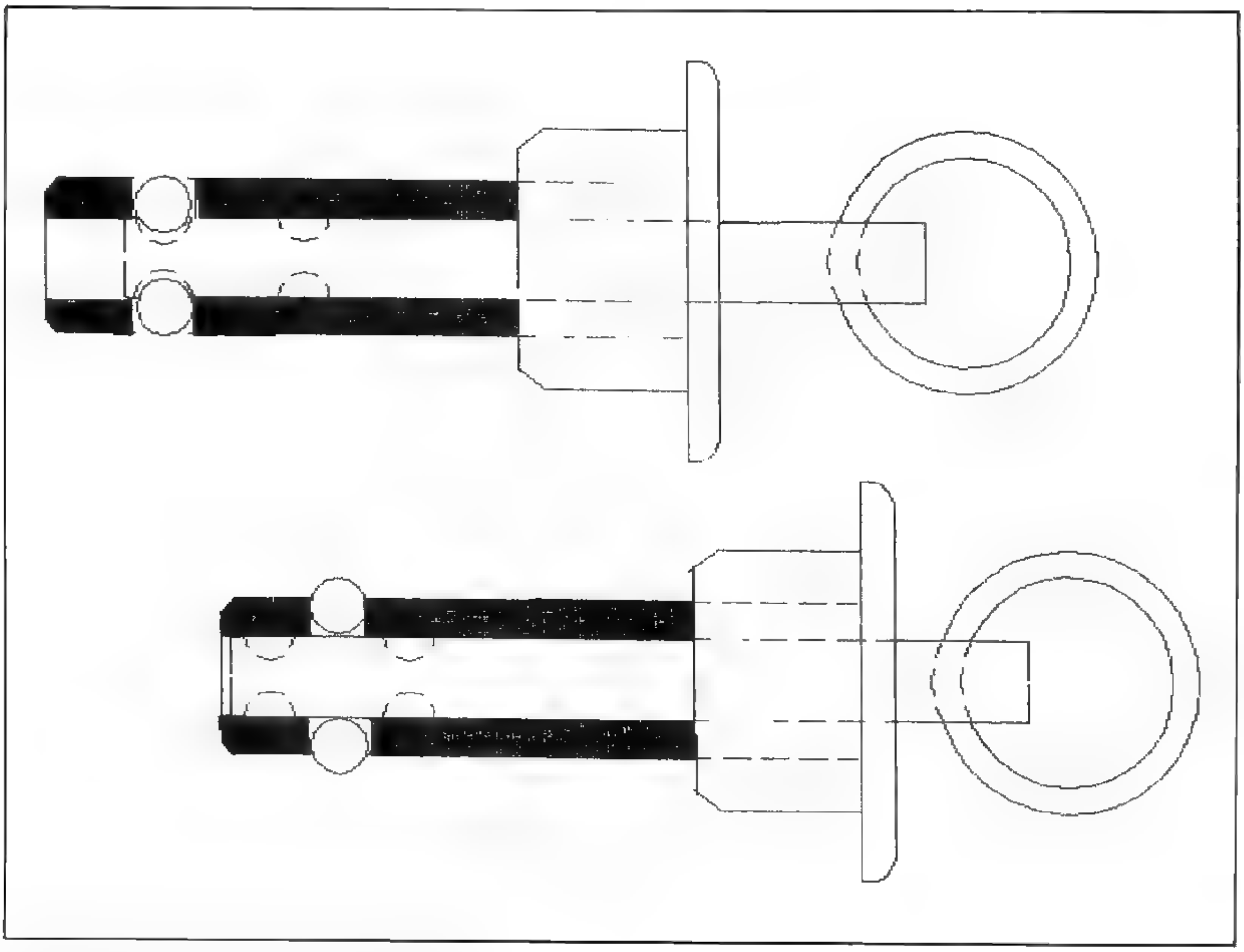
tion came up with the ingenious holder. In their stock configuration, I use a lot of them in the trailer.

Pip pin

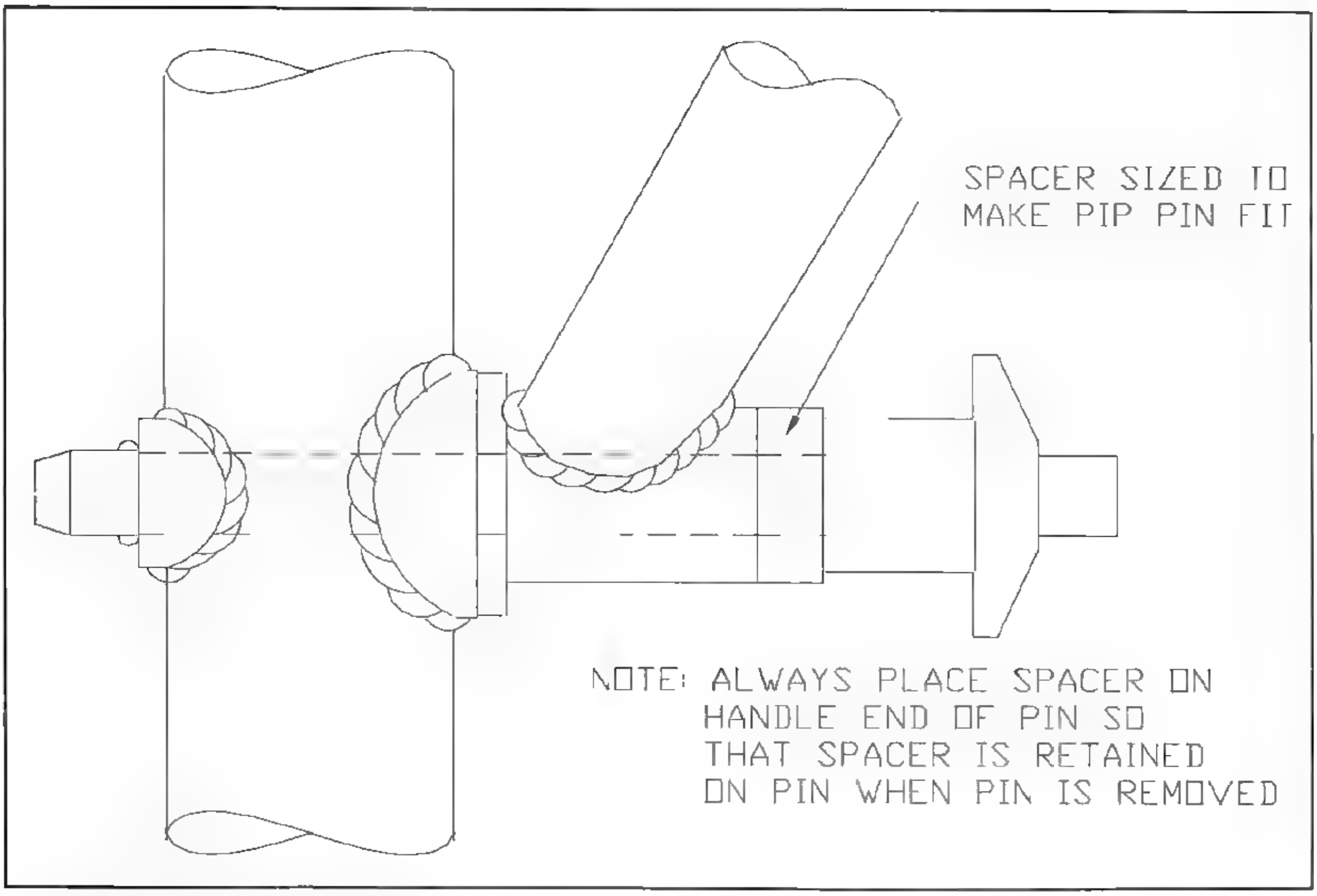
Commonly known as the pip pin, the single acting quick-release pin is among engineering's most convenient and useful items. The basic idea is shown by the cut-away drawing. Two or more balls, captured in bores drilled radially to the axis of the pin, are held in their outward position by an internal plunger—which is concentric to the pin. This plunger is provided with detents so that, when it is displaced in either direction from its normal position, the balls can retract. The plunger is spring loaded to the locked position and its end is provided with either a push button or a pull ring.

In operation, the central pin is either pulled or depressed, the pin is started into the hole, the hole edge pushes the balls into the detents and the pin is pushed through. As the balls clear the far side of the hole, the retractor is released and the balls are forced to their outermost position, preventing the pin from being removed until the unlocking

The family of quick-release pins.



Double-acting ball lock pin (pip pin) cut-away.



Axial spacers to optimize pip pin fit.

plunger is actuated again. Simple and effective, they are available in a wide range of materials, sizes and handle styles.

It seems that each year I find more uses for pip pins. I use them in all sorts of applications, from holding things in place in the trailer to holding the anti-roll bar links in place when I think that it is going to rain. I strongly favor holding front substructures and the like to chassis with pip pins through double shear brackets so that I can replace them in a hurry. Also, I am giving serious thought to using pip pins to hold the spring/shock units in place while testing, just to reduce the downtime while changing springs. (I have always resented downtime while testing, but soaring track rental fees have made it critical.) Some of the homebuilt aircraft people use pip pins to secure the wings to their aircraft.

A pip pin should be installed with the minimum possible amount of both radial and axial play. Limiting radial play is simply a matter of providing a reasonably precise hole diameter. Minimizing axial play often means using a tubular spacer to make up the difference in dimension between the housing and the pin. I place the spacer on the handle end of the pin rather than on the ball end simply because it will then be retained on the pin when the pin is removed and I won't have to go looking for it. We can use washers as spacers so long as they are installed between the handle end of the pin and the work face. They cannot be used on the other end because their I.D. is too large to guarantee that they will stay on.

An intriguing variation on the pin theme is the Expando-Grip quick-release pin from Shur-Lock Corporation, or from the Adjustable Bushing Corporation. This expensive little jewel actually expands to grip the hole wall and positively locks in the expanded position. I have had a sample sitting on my desk for a year and I can think of a lot of uses for this beauty—but they are still a bit rich for my blood.

Push-pull pin

The father of the pip pin was the push-pull pin, a solid shank pin with spring-loaded locking balls that are pushed into the shank when the pin is inserted into (or withdrawn from) its bore—which must be slightly oversize. Pull-out forces are usually in the ten- to thirty-pound range. I have no use for the push-pull pin simply because it is not positively retained.

Finding pins

With the exception of pip pins and AN taper pins, good machine shop supply houses stock most types of pins. The few remaining surplus houses stock many types. The phone company has a pin classification in the yellow pages. Any fastener house can order any type of pin.

That is about all there is to know about pins; that is, it is certainly all that I care to say about them.

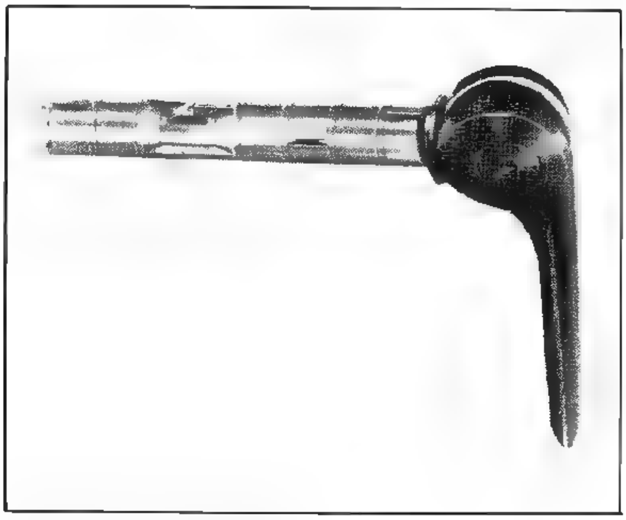
Quarter-turn-to-lock panel fastener

Quarter-turn-to-lock fasteners are used to secure nonstructural panels to frames, as in access panels and body panels. Their only advantage is that they take less time to do and undo than the alternatives. To be perfectly honest, I don't really like any of the quarter-turn-to-lock devices; *something* is always going wrong with the damned things. Like them or not, I use a lot of them. There is simply no other convenient and intelligent way to secure panels—and I deal with a lot of panels.

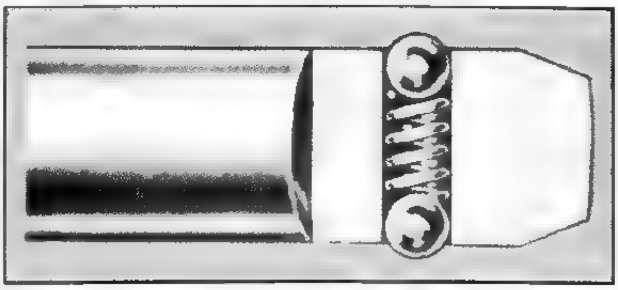
I use Dzus fasteners and Camlocs. In general, I prefer Dzus buttons to Camlocs because the Dzuses tend to be easier to find, more forgiving of minor misalignment and more positive in their locking action. I do not use any of the other quarter-turn devices.

Dzus fastener

So far as I can determine, William Dzus originated the quarter-turn-to-lock panel fastener for use on military aircraft sometime in the 1930s. The Dzus Fastener Company remains the leader in the field. The basic Dzus button assembly consists of a male stud with a screwdriver slot on the top surface, and a spiral cam slot machined into the shank. When the stud is pushed through suitable holes in

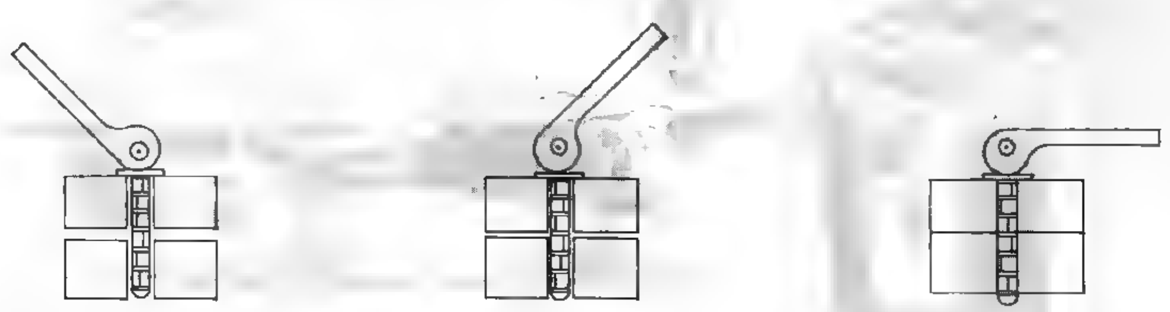


The expanding bushing pin.



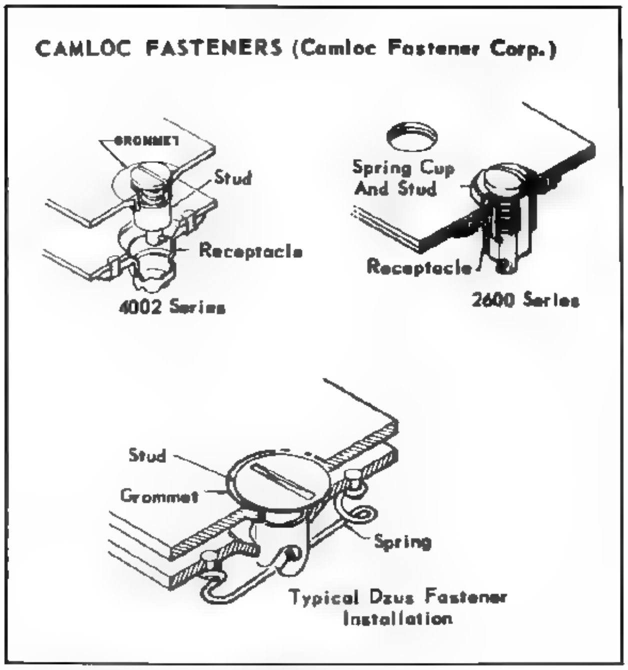
The push-pull pin.

HOW ADJUSTABLE DIAMETER PINS WORK

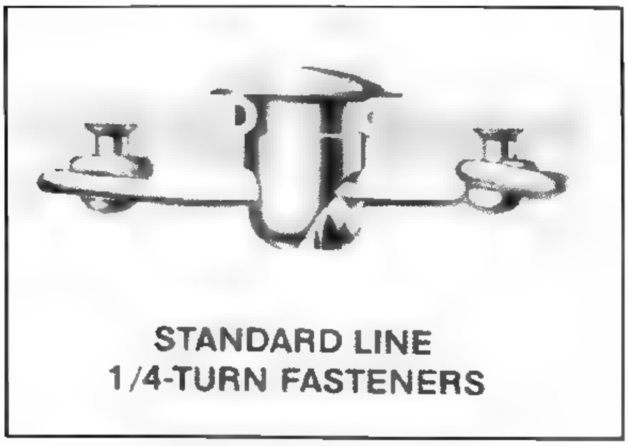


1. Adjustable Diameter Pins, in their relaxed state, provide clearance for free, easy installation and removal
2. Actuation of the cam handle creates an axial compressive force against the segments. This forces the female segments out.
3. The action of the pin assembly provides an extremely tight radial fit and draws the structural members tightly together. The handle is retained in the fully actuated position, since the cam has been rotated slightly over center.

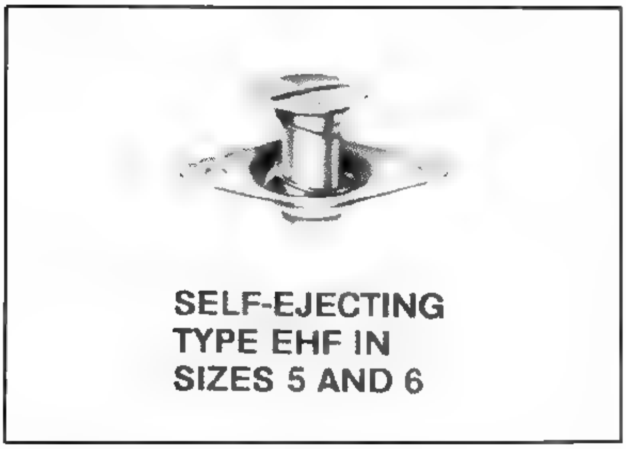
Adjustable diameter pins.



Dzus and Camlock panel fasteners.



The basic Dzus quarter-turn-to-lock panel fastener.



The EFF Dzus self-ejecting panel fastener.

both panels the spiral slot engages a spring wire receptacle which is attached to the fixed panel or frame. Turning the stud pulls the spring wire into the slot. Further turning pulls the wire over a detent in the slot. This locks the assembly under the spring tension. It cannot be released until the stud is pushed down to allow the stud to be turned away from the detent.

The Dzus Fastener Company manufactures a comprehensive line of quick-acting panel fasteners. Their general catalog is eighty-two pages long. With one exception, all of their quarter-turn-to-lock fasteners require special installation tools. Their EFF series of plate mounted fasteners does not, however. The unit is attached to the panel with two rivets. In the uninstalled condition, the EFF stud is almost flush with the underside of the panel, thus making it possible to slide panels into place. They are so easy and convenient to use that it seems silly to use anything else.

It seems equally silly to install either a Dzus button or a Dzus wire onto a fiberglass panel without an aluminum back-up plate. Fiberglass may have many virtues (cost is the only one I can think of offhand), but a bearing material it is not! The back-up plates should have about four times the area of the Dzus plate. They don't have to be very thick (0.031 in. is enough) and they should be dimpled in the center to match the dimple in the Dzus plate. Back-up plates are flush riveted to the inside of the fiberglass panels. This is just one more of the seemingly endless preseason tasks that convert the standard kit car to a practical race car. All of the details pay off during the season, though, when timing is critical.

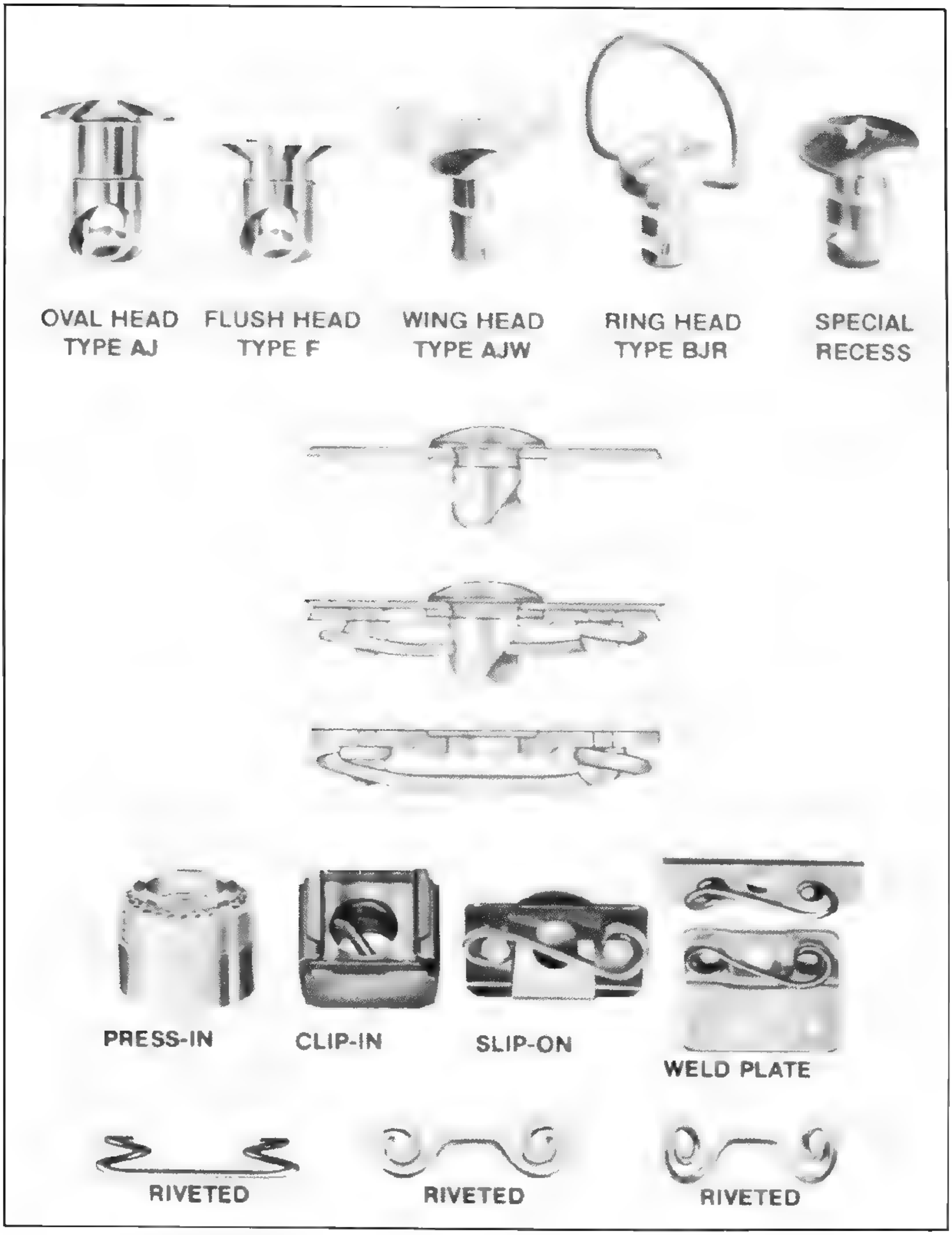
The Dzus wire receptacles are available in graduated depths. I carry several of the inexpensive wires and a few different lengths of expensive studs. The wires are available in several configurations, including an edge mounting clip. The studs are also available in a number of head configurations.

Camloc

The Specialty Fastener Division of the Rexnord Corporation manufactures Camloc quarter-turn panel fasteners. The operating principle is similar to that of the Dzus fastener—except that the function of the components is reversed. A cross pin is inserted in the stud, and the cam is in the receptacle. The spring tension is provided by a coil spring concentric with the stud.

In my opinion, Camlocs are not as secure as Dzuses. The detent in the cam does not seem to be as positive as that in the Dzus stud. I have had Camlocs come loose from vibration even when they have been properly installed and tightened.

Another problem with Camlocs is that the cross pin is only pressed into the stud. If the cross pin shifts while the fastener is installed, you can't get the thing out and you now have a permanently



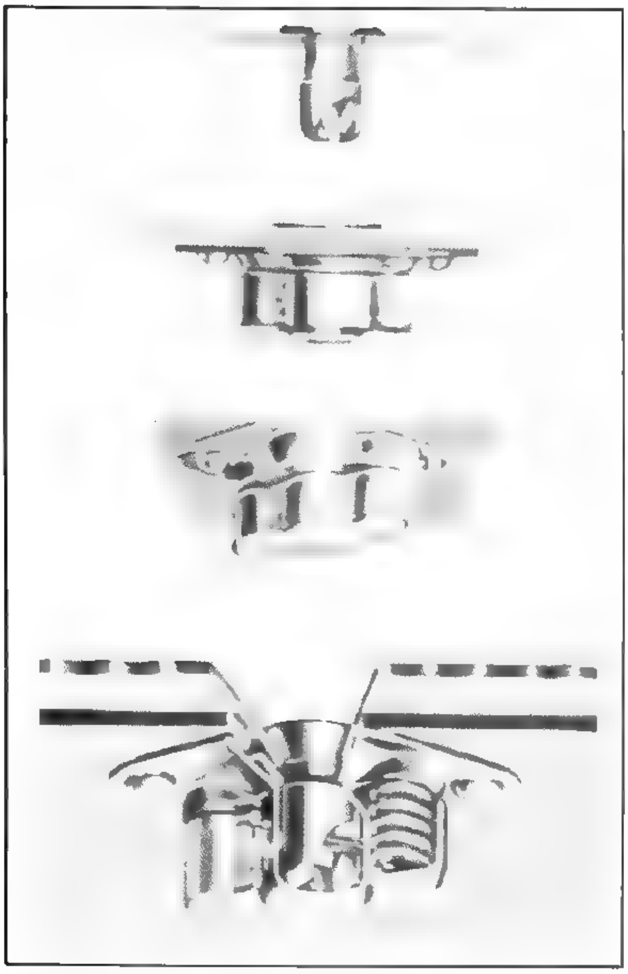
Dzus stud and receptacle configurations.

installed panel. Admittedly, it takes many installations and removal cycles for the cross pin to loosen, but it does happen.

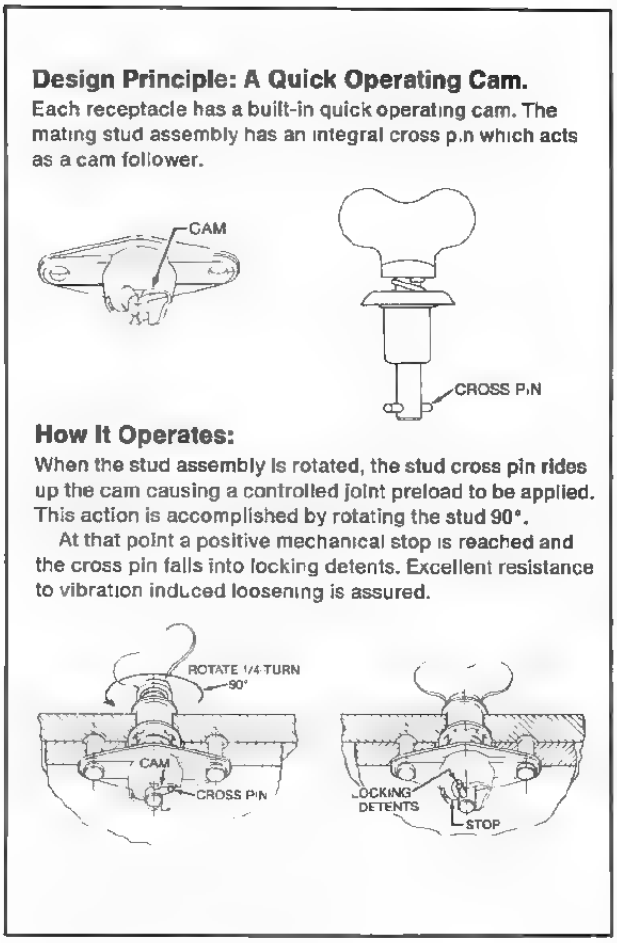
My last complaint is that, even in the relaxed position, the stud extends well below the surface of the panel. Therefore, in order to remove racing car body panels that are secured by Camlocs, the stud and its grommet must be removed from the panel. This means that they must be found before the panel can be replaced. This sounds trivial, but it is not. In the world of motor racing there are few situations more frustrating than not being able to leave the pits because the panel fasteners cannot be found. It is all very well to say that they shouldn't have been lost, but the world doesn't work that way. Murphy lives in the pits!

Push-to-lock fastener

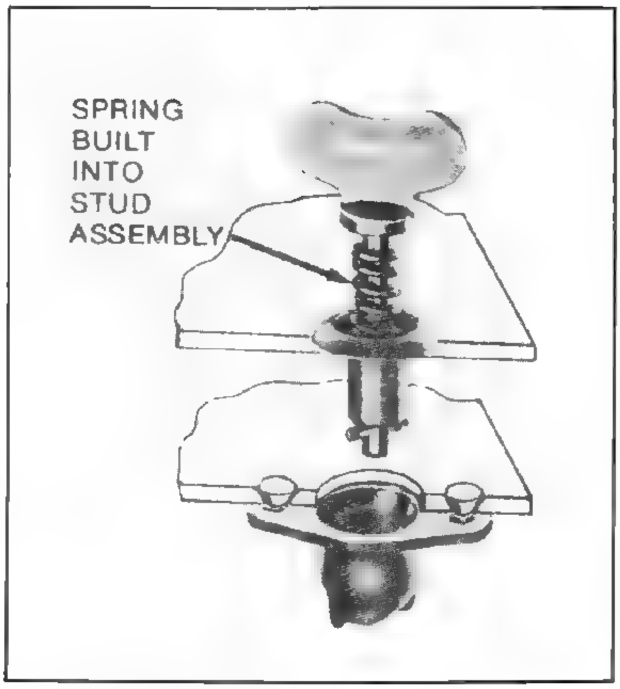
The same companies that make quarter-turn-to-lock fasteners also make push-to-lock panel fasteners. To my mind, these devices are nothing more than deformed nails. They are meant to fasten light access panels onto electronic cabinets. I do not build electronic cabinets, nor have I seen a



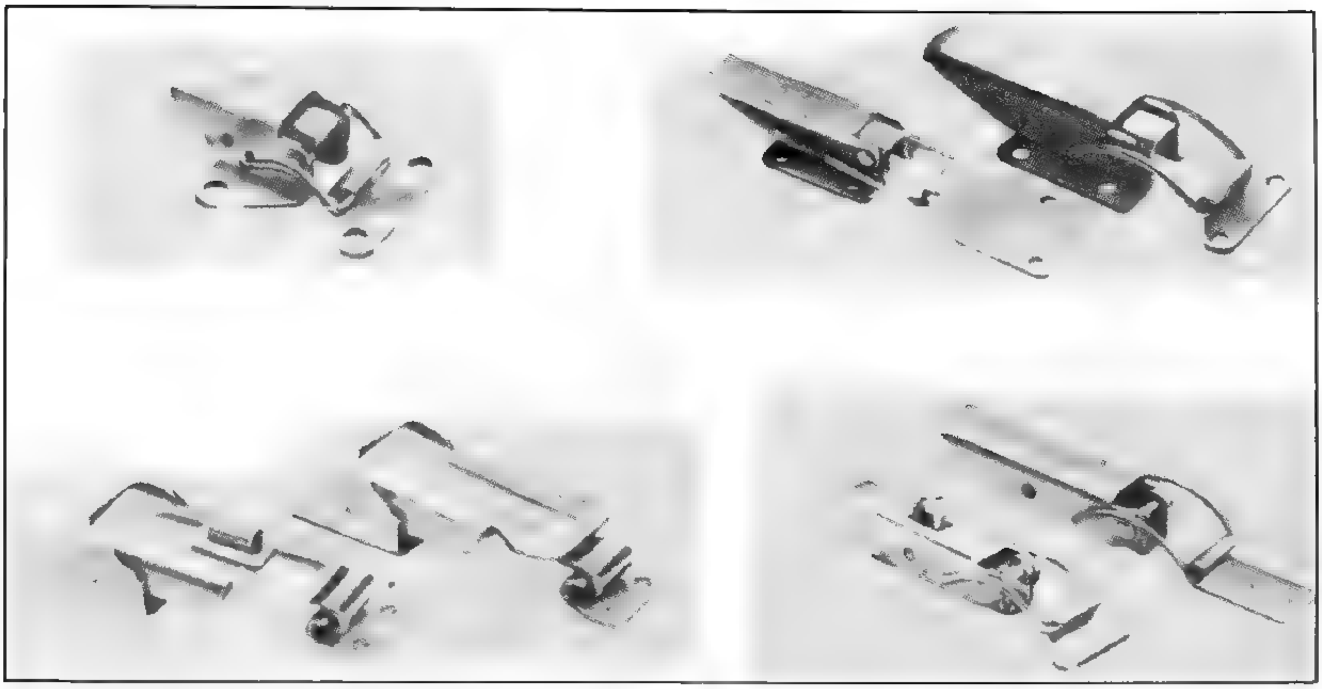
The Dzus quarter-turn-to-lock fastener.



The Camloc panel fastener. Camloc Corporation



The Camlock quarter-turn-to-lock fastener.



Latches by Dzus and Camlock.

push-to-lock fastener that I was willing to use on a racing car. This feeling of mine has done nothing to prevent the manufacturers from doing so, however. I replace the push-to-lock fasteners before I run the car.

Quarter-turn fastener as a panel locator

Much of the trouble that racers experience with quarter-turn fasteners is a direct result of trying to use the silly things as panel locating devices. William Dzus did not have location in mind when he dreamed up the Dzus button. All he wanted to do was to hold the panel shut. If you locate with pins, or guides or lips—or *something*—and use the panel fasteners only to fasten, your life will be more pleasant and your vehicles (or, for that matter, panels or boxes or whatever you may be building) will look a lot better. If racing car manufacturers would do the same, all of our lives would be more pleasant.

Latches

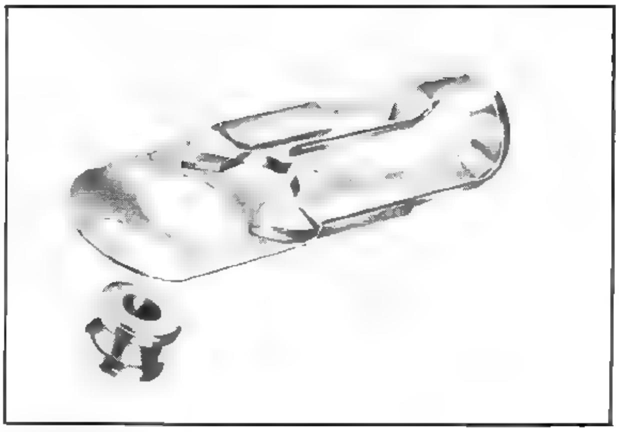
Both Dzus and Rexnord manufacture complete lines of really good overcenter latches and toggle clamps—which I don't use very often but can never find when I do need them.

Dzus makes my favorite latch, the slide latch. This one is aerodynamic, easy to install and positive locking—all in one neat little package. A functionally identical unit is made by the Dimco-Gray Company of Centerville, Ohio (see appendices for address).

Sheet-metal and self-tapping screws

The world of industrial fasteners includes a seemingly endless variety of self-tapping screws. I do not like them or use them. It is perfectly true that self-tapping metal screws and their matching Tinnerman clips have been assigned AN numbers. It is equally true, however, that their use on aircraft is forbidden in the following circumstances: first, as fasteners for the fabrication of primary structure; second, as fasteners for structure or accessories where failure might result in danger or damage to aircraft or personnel; third, where loss would permit the opening of a joint to air or water leakage; and fourth, where they are required to cut their own threads and are subsequently subject to removal.

In other words, aircraft people are not allowed to use them anywhere that they might be important or convenient. If the military and the



Slide latch by Dzus.

FAA place that kind of restriction on the use of sheet-metal screws, it is best to think carefully about using them at all. I am not about to use them on a racing car.

As always, there is an exception. Strangely enough, the exception has to do with racing tires. Occasionally, we run into a situation where either the tire manufacturer has made the beads too large in diameter or the wheel manufacturer has made the rim too small in diameter. When either of these mistakes occurs, the tire can rotate on the rim either under braking or under acceleration. This does terrible things to wheel balance and may result in the loss of air pressure. The solution is to fasten the tire bead to the rim with sheet-metal screws. For this reason and for this reason *only*, I do carry around a supply of sheet-metal screws. I drill a clearance hole in the metal tire rim and screw the self-tappers into the wire of the tire bead.

Remote control cables and linkages

No matter what sort of vehicle we are concerned with, we frequently need to transmit control motions of one sort or another from one part of the vehicle to another. Typical examples include throttle motion from the operator's foot (or hand) to the throttle body, clutch actuation, anti-roll bar adjustment, brake ratio adjustment, aircraft control motions from the stick or yoke to the control surfaces, and boat trim controls. On larger vehicles, like ships and 747s, these control motions are transmitted by fluidics or by synchronous electric motors. On the types of vehicles that I am liable to be concerned with, they are usually transmitted by control cables. There are good control cables and there are less than good control cables. There are also *bad* control cables.

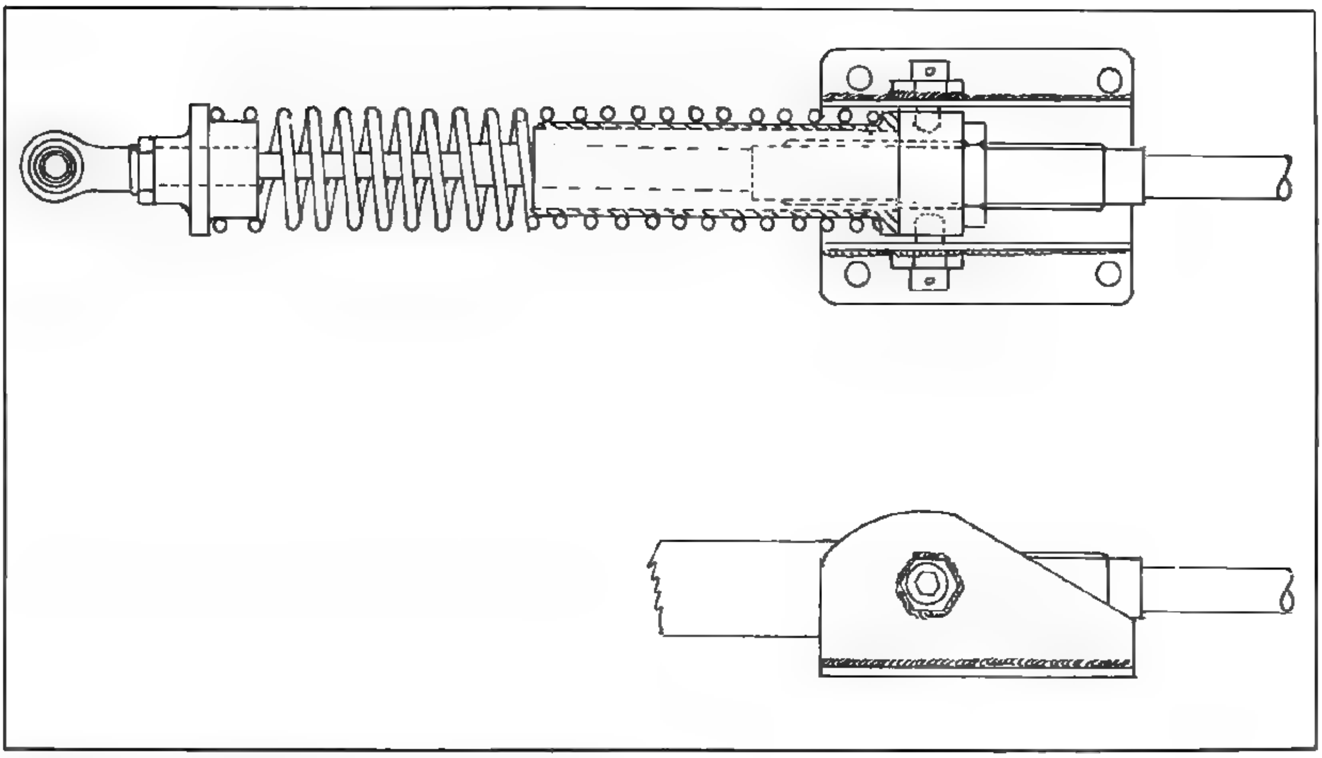
The worst control cables are the familiar motorcycle Bowden cables. They are readily available, easy to make, very cheap and they can get you hurt. I will not allow one on a racing car—period. They are marginally OK for motorcycle and kart throttles, and for the mechanical actuation of brakes and clutches on small bikes. They have absolutely no place on a racing car, an aircraft or a boat. They stretch, they jam and they break—and Murphy makes sure that they do so at the wrong time. But then again, there is no *right* time for this sort of thing.

I am not going to get into aircraft control cables, pullies and systems. Those who are allowed to play with them know what they are doing. Even if they do not, each system must be signed off by an FAA inspector before it can be flown. The inspectors do know what they are doing. I will, however, mention in passing that I see no reason for me to reinvent the wheel, and so when I need to operate ramps and lifts in racing car transporters, I use the aircraft cables and go by the book in designing and installing the systems. The book in this case is *Aircraft Maintenance Manual*, CAM 18, from the Experimental Aircraft Association (see appendices for address).

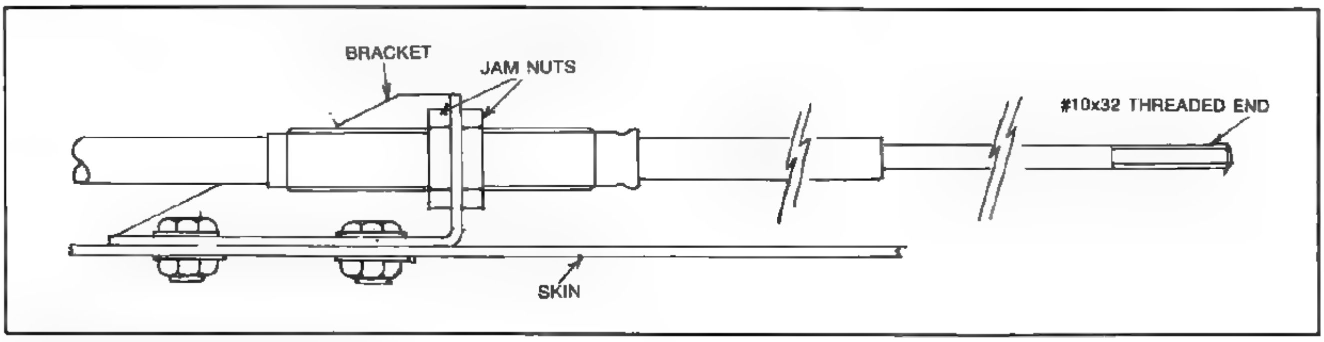
With either cars or boats, control cables should be of the push-pull variety that will function both in tension and compression. This allows several good things to happen. For instance, the driver can pull the throttle back with his foot if the return mechanism fails, and anti-roll bars and boat trims can be moved in either direction. Try any of the above with a Bowden cable!

All of the push-pull cables are similar in design and construction. They are made up of an inner

Bracket for threaded bulkhead-type push-pull control cable attachment, and hose clamp bracket for slotted groove-type push-pull cable attachment.



Compression spring throttle cable abutment with swivel mount.



Attachment of throttle cable to chassis.

operating member, usually a spiral-wound steel cable, a plastic antifriction liner, sometimes nylon and sometimes Teflon, a spiral steel conduit or housing and a plastic cover. The ends of the housing terminate in either threaded or grooved end fittings while the ends of the inner member terminate in ball-and-socket swivel joints from which extend male threaded ends. The swivel ends will typically accept from 14 to 18 degrees of angular misalignment—which is enough to allow you to build just about any sort of linkage without getting things into a bind. Strokes vary between 2 and 6 in.

The end terminals are selected to allow either bulkhead mounting with jam nuts or grooved swivel mounting for use with a clamp. The bulkhead system has the notable advantage of easy adjustability, while the clamp is both easy and economical to mount. I use both configurations, depending on which seems more suitable for what I am doing. The ends are also available in a variety of locking and nonlocking knobs and T-handles.

On racing cars, we use push-pull cables as throttle cables and as anti-roll bar adjusters. On boats they are used for throttle cables, steering cables and trim tab actuators. On trucks they are split-axle shift actuators, fuel tank selectors and so on. At the operating (and operated) ends you can use either female clevises or female rod end bearings. I prefer the rod end bearings because of their lack of friction and their self-aligning nature. I usually use the Heim Manufacturing Company's bronze outer race bearings because they are self-

lubricating, grit insensitive and cheap. The loads involved are usually low enough that a light-duty rod end is plenty good enough. I do not, however, use Nippon Miniature Bearings' (NMB) Mohawk series—even here.

The best of the push-pull cables is manufactured by Cablecraft, Inc. It is far and away the most flexible of the group, with 2 in. minimum bend radius in the #10–32 size as compared to 4 in. for the best of the rest. It has less frictional drag than the rest and is very heat resistant. Of equal importance, Cablecraft has a nationwide network of distributors who make up the cables to order in literally a matter of minutes. Cablecraft makes two series of control cables; you want the one with the green outer cover.

American Chain and Cable (ACCO) units are comparable in heat resistance, frictional characteristics and quality to Cablecraft's, but are a long way behind in flexibility. They are the second best cables available, and are almost impossible to obtain. Morse units are of excellent quality as well, but tend to bind when exposed to heat. These cables are the easiest of all of the quality units to find, as virtually every boat dealer in the western hemisphere stocks a reasonable selection.

The basic design principles remain pretty constant for all cable-operated devices and linkages. This being the case, rather than going through a series of detailed designs and installations, I am going to describe in some detail how I install the most critical of the cable systems and the throttle linkage, and let you carry on from there.

At first glance, the design of a throttle linkage seems about as simple as anything mechanical ever gets. Anyone who has ever had a throttle linkage either jam wide open or break will dispute this point. They will also tell you just how crucial good design and first-class components are to the continued health and happiness of the driver. The basic requirements for a good throttle linkage are simple: first, to transmit the motion applied to the throttle pedal or lever by the foot (or hand) of the driver to the actual throttle device. Second, to do this smoothly with a minimum of friction and *no* stickiness or notchiness at all—despite the best efforts of a hot and dust-filled environment to jam up the works. Third, to ensure that the throttle closes—instantaneously—every time the driver removes their hand or foot from the control. Fourth, to provide the desired relationship between pedal movement and throttle opening. Fifth, to be so designed and installed that the possibility of accidental damage (by stepping on the cable, or by jamming it between parts while changing engines) is minimized. And sixth, to conveniently provide the required degree of adjustability.

The realization of these requirements may or may not be simple. Assuming that the device is to be linear—or nearly so—in operation (that is, the throttle opening will increase by the same amount for each increment of pedal travel), the first task is to lay out the desired amount of pedal travel versus the required amount of throttle displacement from idle to wide-open throttle. The next step is to position the ends of the cable at the proper distance from their respective pivots to achieve that relationship.

The third step is to find a strong, convenient and accessible mounting place for both the cockpit and the throttle body ends of the cable. I usually use the threaded bulkhead-type mounts because they give me more latitude for adjustment. I make my own mounting brackets, and I gusset the ears so that the bracket will not fail at the obvious stress raiser. The completed bracket is illustrated here. I bolt the bracket to the chassis rather than riveting it so that I can get it off when I have to. An alternative, at least for tube frame cars, is the hose clamp bracket.

The operating end of the throttle linkage can be attached either to the chassis or to the engine. Even with rigidly mounted engines, there is liable to be a small amount of relative motion between the engine and the chassis. I therefore prefer to mount the throttle cable bracket directly to the engine so that any rocking of the engine in its mounts will not affect the position of the throttle.

Since no two engines are dimensionally exact, and since pedal position has to be varied to suit individual drivers, some sort of convenient provision must be made to adjust the position of the throttle pedal. I prefer to install a left- and right-hand threaded connector at one end of the assembly or the other. Because I have seen Super Mechanic tear the threads out of aluminum connectors, I usually make mine from steel. If I am being particularly weight conscious, I make them from 7075-T6 aluminum hex bar stock and replace them whenever the threads begin to look even a little bit worn.

The last step is to lay out the run of the cable. It is important to make sure that it cannot be pinched, crushed or stepped on, and that it stays well away from heat sources like exhaust systems, water pipes and oil lines. It is also important to make sure that it is well out of the way of the other controls.

Note: The design of the actual throttle linkage itself is outside the scope of this book. I touched on it in *Prepare to Win* and will cover it in depth in *Design to Win.*

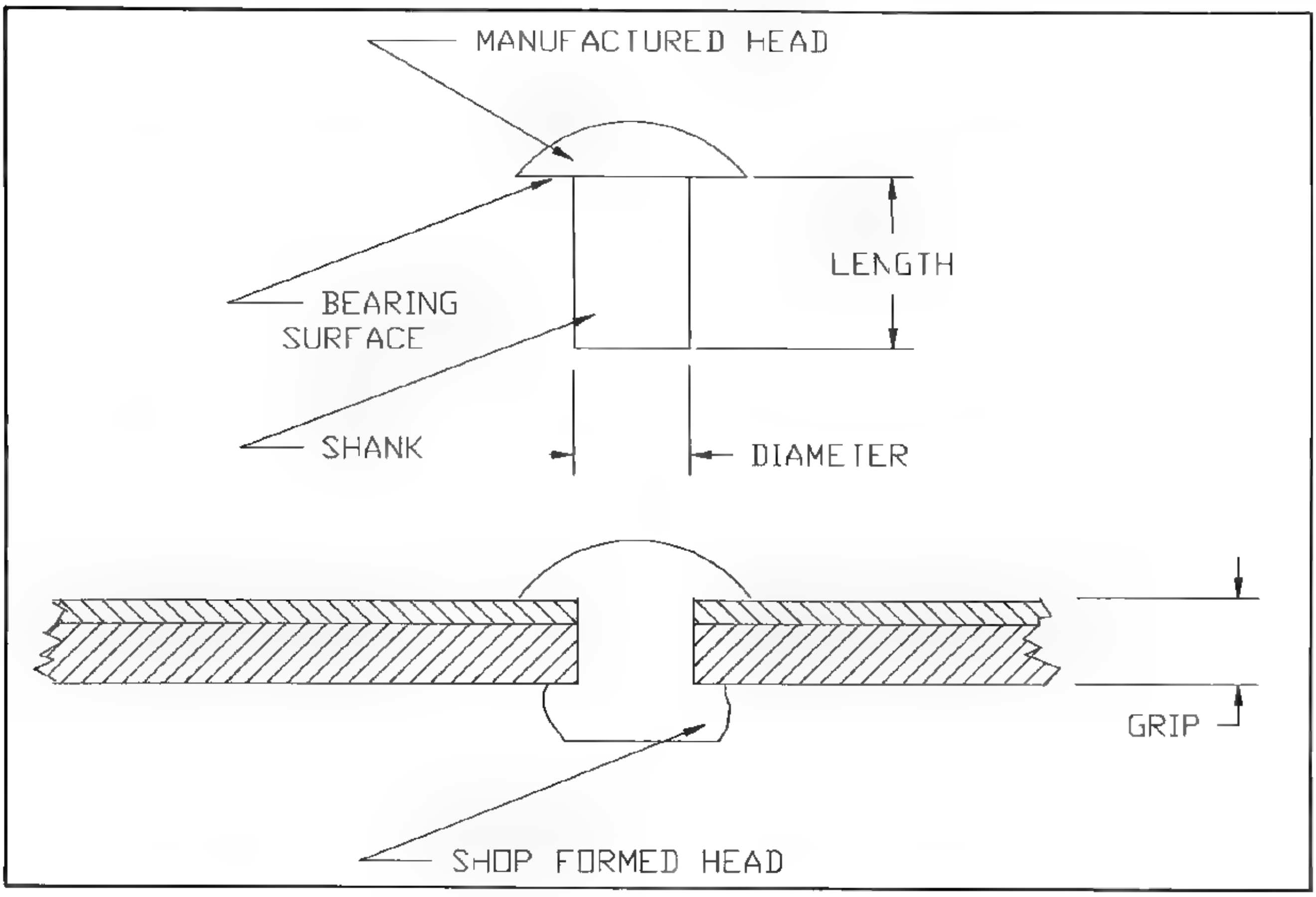
9

Rivets and riveting

The rivet is a simple device that has been around since before history began. It is manufactured in the basic shape of a mushroom from any of several relatively soft metals. The cap of the mushroom is termed the shop-formed head of the rivet. The stem is termed either the stem or the shank. In use, the rivet stem is inserted through predrilled holes in two or more layers of material until the head bears against one surface of the work. The protruding stem is then hammered into a matching (shop-formed) head on the opposite side of the work and thus the workpieces are fastened or joined together.

The origins of the screw thread are caught up in the excitement and romance of the Daedalus legend. No such aura surrounds the lowly rivet. Not only does no one know when and where the use of the rivet began, so far as I have been able to discover, no one cares. After all, at least until very recent developments in aerospace, the rivet has been a strictly utilitarian device, lacking both elegance and class. Its inspiration came, not from the structured beauty of the chambered nautilus, but from a toadstool.

Throughout the centuries, what the rivet has lacked in glamor and mystery it has made up in utility. In the Bronze Age rivets were used to hold the handles on various things from swords to sickles. At the beginning of the Iron Age they were



The basics of the rivet.

Ultimate shear strength of various types of rivets

Rivet type	Solid core blind		Hollow core blind		Solid core blind		Solid core blind		Lock stem		Solid (AD)	
Manufacturer	Avdell (Avex)		Cherry N		Cherry Q		Cherry Q		Cherry Max		Various	
Rivet material	5056 Alum		5056 Alum		5056 Alum		Mild steel		5056 Alum		2117 Alum	
Stem material	Carbon stl		Carbon stl		Carbon stl		Carbon stl		4130 Stl		None	
Rivet diam. (in.)	1/8	5/32	1/8	5/32	1/8	5/32	1/8	5/32	1/8	5/32	1/8	5/32
Ultimate shear strength (lb.)	210	305	200	325	350	525	550	800	615	976	388	593

Table of comparative rivet strengths.

commonly used to join the plates of the body armor that protected the wealthy in battle, and to fasten the handles or grips of knives, swords and the like to the tangs forged onto their blades. From the early years of the Industrial Revolution until very recent times, hot rivets were used to join iron/steel plates (as in boiler, ship and tank) as well as beams and girders (as in bridge and building). The reason was simple—a steel rivet, driven red-hot, is guaranteed to do the following: first, completely fill the hole through which it is driven, thus doing away with the possibility of a loose fit and later trouble. Second, contract upon cooling, thus generating a residual tension stress that will both clamp the joined parts and prevent later loosening of the fastener. Third, be considerably cheaper, faster to install and more idiot-proof than a nut and bolt. And finally, provide hours of utter fascination to anyone fortunate enough to have seen the Mohawk high-steel workers tossing hot rivets from forge to riveter while balanced on I-beams hundreds of feet above the ground. It was a lot better than watching jai alai.

Rivets still hold the handles to the tangs of our better knives and, for that matter, to most of our pots and pans. But these days they do a lot more than that. For instance, they join the majority of the sheet-metal panels and stampings that make up most aircraft structure—and the majority of our racing car chassis. Almost all of the riveted ships, boilers and the like have disappeared from service, but a great many of our older commercial buildings and bridges are still of riveted construction, are doing fine and will presumably continue to do so for a long time to come.

While modern welding techniques and equipment have pretty much spelled the end of the construction hot rivet, the increasing importance of sheet-metal and thin-plate construction in aerospace (and in racing cars) as well is in truck bodies, lawn furniture and the like has led to the development of rivets and riveting techniques that have created a whole new science of joining. In thin-metal construction, riveting offers several advantages over welding, the use of threaded fasteners or adhesive bonding.

Compared to welding, riveting causes no dimensional distortion and no alterations in the grain structure or heat treat properties of the metal to be joined. Rivets permit the joining of dissimilar materials and allow joined components to be disassembled with relative ease.

Compared to the use of threaded fasteners, riveting is faster, cheaper, lighter, presents a smoother finished surface and, in thin sections, is at least as strong. Further, since no torque is applied during the installation of a rivet, there is no residual shear stress to relax and allow loosening of the fastener—after it has been installed, of course. There is also no residual tension stress to provide clamping force. So rivets are not very strong in tension.

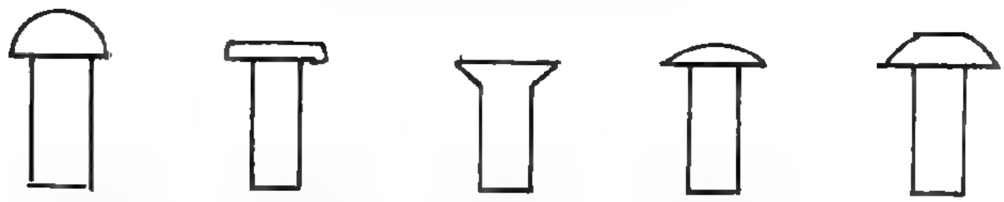
Compared to adhesive bonding, riveting requires less surface preparation, lower levels of sanitary working conditions, less complex and less expensive installation equipment, and a lower level of skill on the part of the work force. After installation, the joint (as well as the joined components) can be more easily inspected and disassembled.

Until we get into advanced aerospace applications, rivets and riveting equipment are readily available and cheap. The skill level required for installation is minimal and the joins, assuming correct design, are more than satisfactory. However, the advent of the monocoque race car chassis and of load bearing sheet-metal panels in most tube-framed cars has increased both the scope and the complexity of riveting on racing cars. The days when a fistful of hardware store pop rivets and a 1/8 in. drill would get us by are gone forever. We will first discuss the rivets themselves and then the design of the riveted joint.

Types of rivets

There are several different types of rivets. They range from the eyelets through which we pass

Hard rivet identification



Round head MS20435 AN–430 AN–435

Flat head AN–442 AN–441

Counter-sunk head MS20426 AN–426

Brazier head AN–455 AN–456

Universal head MS20470 AN–470

Rivet code number: AN470AD5–8

AN 470 AD 5 8

- 8 — Length in $\frac{1}{16}$ in. ($\frac{8}{16}$ in. – $\frac{1}{2}$ in.)
- 5 — Diameter in $\frac{1}{32}$ in. ($\frac{5}{32}$ in.)
- AD — Material code (2117 Aluminum)
- 470 — Head style (Universal head)
- AN — Specification

Material	Head marking	Material code	Notes
1100 Alum	None	A	Weak but can be welded after set
2117 Alum	Single dimple	AD	Standard airframe hard rivet
2017 Alum	Raised teat	D	Icebox rivet
2024 Alum	Raised double dash	DD	Icebox rivet
5056 Alum	Raised cross	B	Obsolete—similar to AD rivet
Mild steel	Recessed triangle	None	Cadmium or zinc plated
Stainless	Recessed dash	F	
Monel	Plain, twin teats or dimples	M	
Copper	Plain	CU	

Note: Round head, flat head and brazier head rivets are obsolete.

AN rivet identification code.

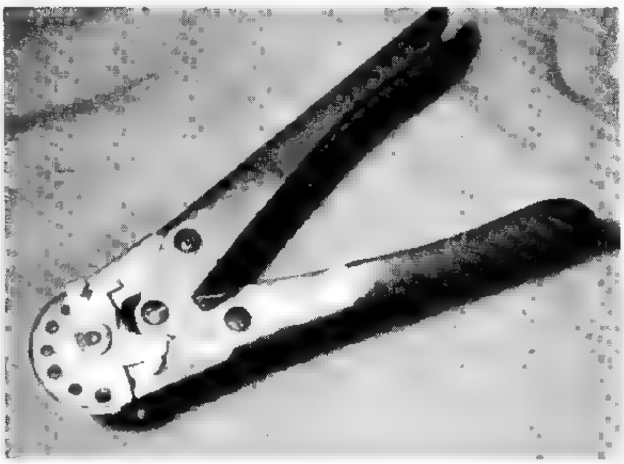
shoelaces to some exotic titanium devices that hold together hypersonic military aircraft. The rivets that directly interest us lie somewhere in the middle of the range; we are going to ignore the low end of the rivet scale completely. I will touch on the high-tech end briefly at the end of the discussion.

Rivets can be divided into two broad groups. Solid or hard rivets require access to both sides of the work and are installed by upsetting the shank into a shop-formed head with some sort of hammer. Blind rivets are tubular in construction and are installed from one side of the work only. They are upset by pulling a mandrel into the tubular rivet with some sort of special tool. Popular opinion has it that solid rivets are much stronger than blind rivets. This opinion is a holdover from the days when the only available blind rivets were what we call trim or nonstructural rivets. The table shown here tells us that the opinion is now out of date—even if the nonstructural blind rivets are not. Since riveting began with the solid rivet, so will I.

Solid rivet

The traditional aircraft and race car rivet is the hard or solid or buck rivet. It is stronger in both tension and shear than most blind rivets. It is dirt cheap (on the surplus market) and it is very good looking. The solid rivet is also a good bit more work to install, requires access to both sides of the work and the use of a pneumatic riveting hammer with the correct rivet sets and bucking bars. Most racing chassis used to be hard riveted. This was because the extra labor was more than offset by the low cost of the rivets and the quickness of the riveters. They were *fast*—and this made hard riveting in production a pretty economical process.

Few tubs are hard riveted today. For one thing, there aren't that many skilled riveters around anymore. For another, the cost of labor has reached the point where it is more economical to employ relatively unskilled people with expensive but foolproof tooling and rivets, than it is to employ expensive people and cheap rivets. Lastly, it has dawned upon most of us that the better blind rivets do every bit as good a job of holding things together.



Hard rivet cutter.

There are (or at least used to be) several different materials used for aircraft hard rivets—four different aluminum alloys plus mild steel, stainless, Monel and copper. In order the reduce confusion and prevent disaster, the rivet material was (and is) coded into the rivet heads in the AN version of Braille, as shown. The picture also illustrates the styles of shop-formed heads that are available. We are interested only in the brazier and universal heads (which I consider to be interchangeable) and in the countersunk head.

We are not interested in Monel, steel or copper hard rivets. We also do not want to know about 2024 or 2017 (D or DD) rivets because these are heat treated to the T-6 condition in manufacture and then kept from age-hardening by being stored in refrigerators at very low temperatures until just before use. After they are driven, they age-harden all by themselves to high strength. Cold storage delays the age-hardening process, it does not arrest it, and the rivets will remain driveable for only a limited time. Over-age refrigerator rivets often appear on the surplus market and must be avoided. They will crack or split if you try to buck them.

I have not seen a 5056 (B) solid rivet in years. They are also set in the as-received condition but are (or were) not as strong as the AD, so we have no need to know about them.

The 1100 (A) rivet is made from commercially pure aluminum, is dead soft and is usually set with a hand-held hammer. It is quite useful in the fabrication trade because it is both very malleable and weldable. We use it to hold baffles in oil reservoirs, water headers and fuel tanks, and then weld around the rivet to form an effective seal. It should not be used for structure because, being dead soft, it is also dead weak (about 16,000 psi). I use the 1100 rivet in 3⁄32 in. diameter, 100 degree countersunk configuration for the trailing edges of wings because I can squeeze them with my favorite lever clamp (more about that later) without distorting the thin sheet metal.

The only hard rivet that we should use for structure is the 2117-T rivet called the AD rivet. It is mildly heat treated (to the T-4 condition or about 38,000 psi) in manufacture. This is a partial heat treatment, which allows the rivet to be bucked or upset. The rivet work hardens to full strength as it is bucked. It was once called the field rivet simply because it could be used in the field as opposed to in the factory. The AD rivet is the universal race car chassis and structure hard rivet.

The bottom line of buck rivet identification is simple: if the rivet has a protruding head marking—

or *any* head marking other than a single dimple, don't use it. If a rivet has no head marking at all, it is dead soft and is suitable for special purposes only.

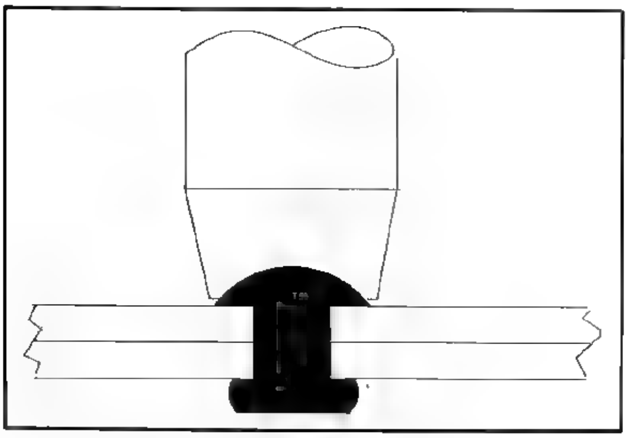
Buck riveting technique

Practice bucking rivets on scrap material before attacking anything valuable. Two people do it a lot better than one. Use the correct length rivets; the rivet should be between 1 and 1½ diameters longer than the work thickness. Solid rivets can be cut to length with diagonal cutting pliers and then filed square, but it is a pain. If they are not finished square, they cannot be properly upset. It is much easier to use the made-for-the-job rivet cutter. These are available from your local aircraft supply store or from Aircraft Spruce. I carry a rivet cutter and several long AD rivets around with me.

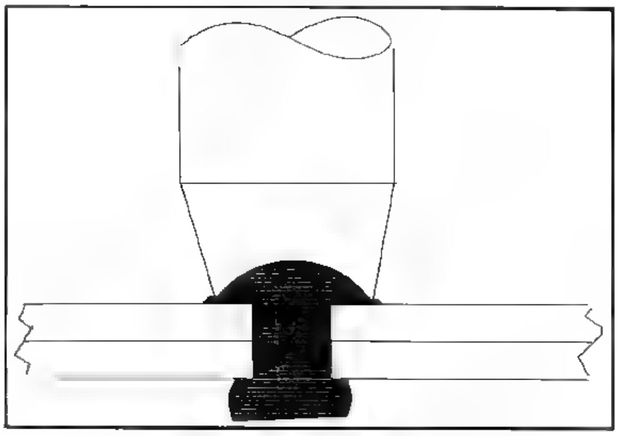
To upset the rivet, you hold a bucking bar against the protruding stem on the far side of the work and hammer the manufactured head with a special pneumatic hammer. The force of the hammer blows is transmitted through the rivet shank, and the shop-formed head is upset by the bucking bar. Do not attempt to use an air chisel or air hammer—they are far too violent and are not noticeably controllable. Proper rivet bucking guns are available for around $100 from Aircraft Tool Supply in Michigan or from any aircraft tool house.

An air pressure regulating valve on the rivet gun is a necessity. The tool that actually contacts the manufactured head of the rivet is called the rivet

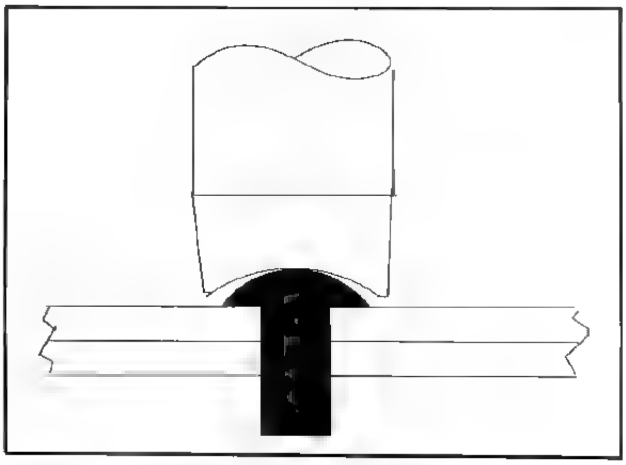
Bucking rivets.



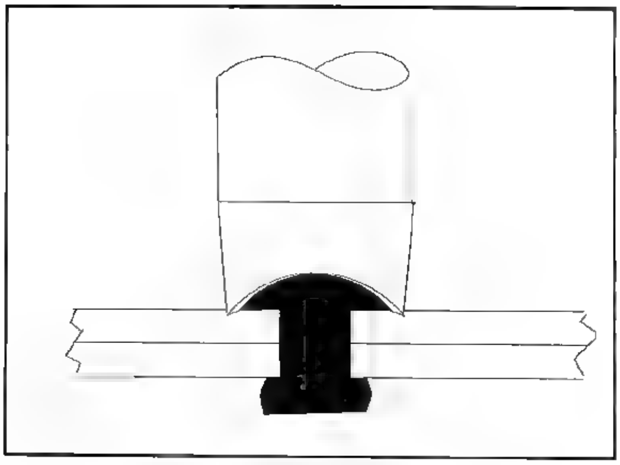
Possible errors in hard riveting: Correct rivet set.



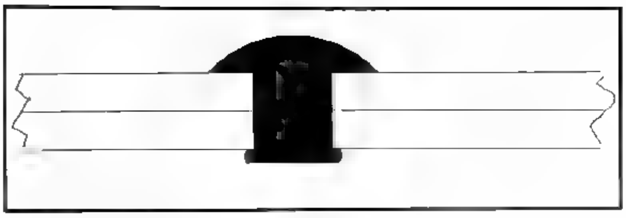
Rivet set too small.



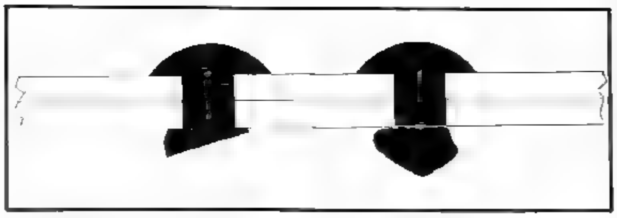
Rivet set too large. Shown before bucking.



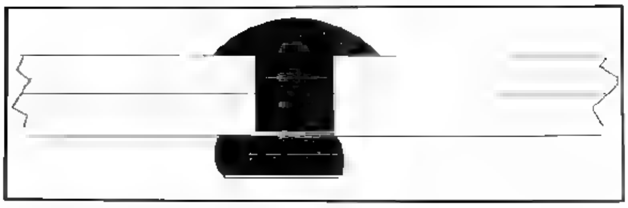
Rivet set too large. Shown after bucking.



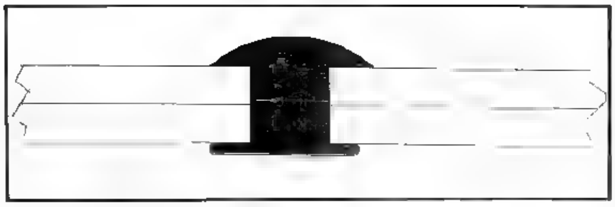
Rivet too short.



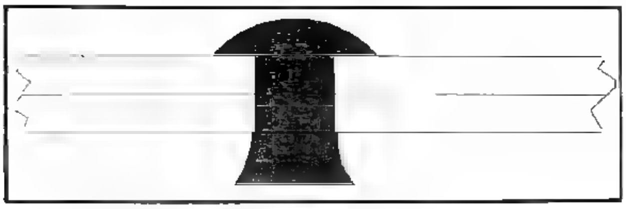
Bucking bar not held square.



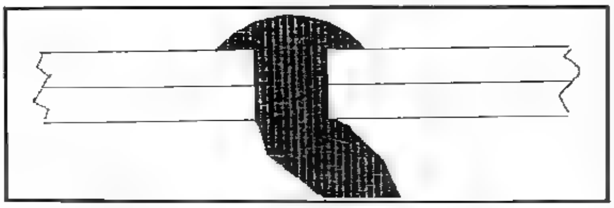
Shop-formed heat offset.



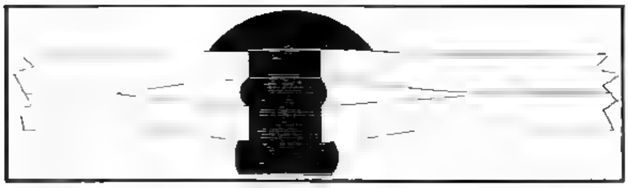
Rivet hit too hard or too long.



Rivet not hit hard enough.



Rivet too long.



Sheets not pulled together.

set. It is removable from the bucking gun. The rivet set must fit the rivet head properly or you will either flatten the rivet head or leave nasty raised marks called eyebrows on the sheet metal. This means that you are going to need several rivet sets. The bucking bar should be hefty and, again, you will need several. If you are going to buck rivets, you are first going to spend some money for equipment.

Push the rivet firmly into the hole with the rivet set, hold the bucking bar hard against the rivet and pull the trigger. The photo and drawings illustrate the basic ideas. Adjust the air pressure until a single burst of ten to twelve blows correctly upset the rivet. A fair amount of practice is necessary to achieve proficiency. Once proficiency *is* achieved, however, the speed with which rivets can be bucked is astounding. It is necessary to inspect both sides of each rivet after bucking. If an AD rivet is not upset sufficiently, rebuck it—soon. Most other deficiencies are cause for removal and replacement. The common shortcomings are shown in this illustration. Some tips are:

- When drilling in the vicinity of fuel cells, water tubes, fluid lines and so on, and removal is not practical, install a positive stop on the drill bit. This will prevent drilling a neat, round hole in the fuel cell, water line or whatever. You can use anything from masking tape to a real drill stop from your aircraft tool supply house, but use *something.*
- All rivet holes must be deburred, on both sides of the sheet. Otherwise either the rivet head will not sit flush with the worksheet or the two worksheets will be held apart.
- All rivet heads inside of fuel cell cavities must be covered with tape to prevent chafe damage to the cell. It is lighter (and easier) to place the manufactured head inside the fuel cell container and form the shop head outside. Make very certain that all of the swarf is cleaned out of the cavity before installing the cell.
- Masking tape over the manufactured head will hold the rivets in place as you go and will minimize the skin damage when the rivet set slips. This allows a whole row (or rows) of rivets to be preinstalled and then bucked in sequence.
- Make sure that the person with the bucking bar is backing up the same rivet that the gun operator is about to shoot.

• Start from the center of the work or rivet row and work outward toward both ends.

Blind rivet

The basic way in which all blind rivets work is illustrated by these drawings. The rivet itself is a metal tube with one end upset to form the manufactured head. The rivets are formed from various materials, the most common of which are aluminum alloy 5056, low-carbon steel and Monel. A mandrel (often called the stem or the nail) with a bulbed or upset head is installed through the rivet during manufacture.

In use, the rivet mandrel is inserted into the jaws of a special rivet tool. The rivet is then pushed through sized holes drilled in the parts to be joined until the manufactured head bears against the top surface of the work. The tool then pulls the bulbed end of the stem up into the rivet tube. The bulb expands the tube outward, forming the shop-formed head on the far side of the work. At the same time, the contraction of the rivet draws the sheets together. Depending on the design and purpose of the rivet, there are four different options to finish the job: first, the mandrel can be pulled right through the rivet, leaving a tubular rivet of limited shear strength and resistance to fatigue.

Second, the mandrel can be manufactured with a notch just about the upset head so that the mandrel will break off when the internal stress reaches a predetermined level, leaving the bulb captured by the shop-formed head of the rivet. This is the most common method. No part of the rivet stem remains in the shear plane and the result is, in effect, still a tubular rivet of limited shear strength and fatigue resistance.

Third, the mandrel can be manufactured with a notch located so that it will break above the work surface. In this case the mandrel will have to be cut off and trimmed flush in a separate operation, but because the mandrel is tightly squeezed by the rivet tube and positively retained in the shear plane, the installed rivet is immeasurably stronger in shear and equally superior in resistance to fatigue.

And fourth, the rivet can be cleverly designed so that the stem breaks at or just below the top of the manufactured head. This provides the shear strength and fatigue resistance of method three, while doing away with the necessity for the separate cutting and trimming operations. In this case, selection of rivet length with relation to the work thickness is even more critical than usual.

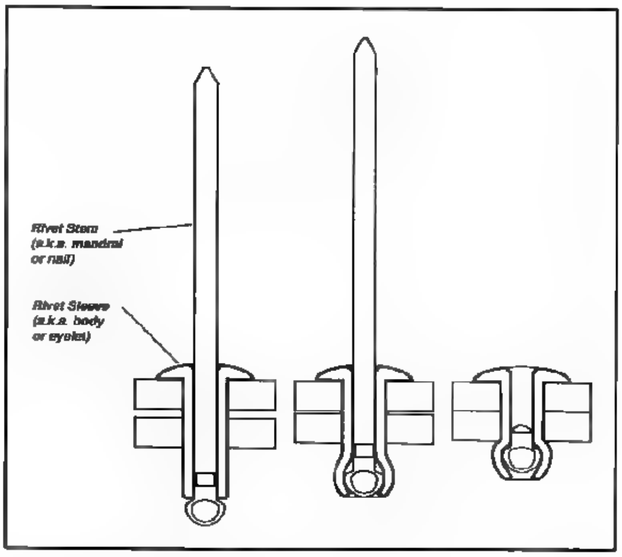
Nonstructural blind rivet

In all probability, most of the rivets that you are liable to use are blind rivets, and most will be of the garden-variety hardware store pop rivet. This is convenient because they are available in blister packs of five and ten at every discount and hardware store. It is not very smart, however. Hardware store rivets are not very strong and those of my friends who have used them for anything more critical than trim repair have eventually regretted it. I don't even keep the things in the shop, for reasons relating to both price and quality.

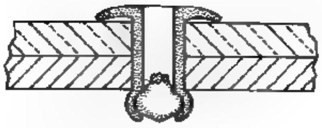
The major reason is that I don't want to have to stock several different lengths of pop rivets. All conventional rivets are grip length critical—the length of the rivet relative to the thickness of the work is almost a fixed dimension. The rule of thumb is that the length of the rivet under the head should be 0.9 to 1.4 times the thickness of the work. If the rivet is too short, there is not enough material to form a satisfactory blind side (shop-formed) head. If it is too long it won't upset properly either, and will look messy to boot. The critical nature of rivet length combined with the various thicknesses of materials to be joined means stocking a bunch of different length rivets—which tends to be a pain.

The Avdell Corporation has a device called the Avex rivet which neatly solves this problem. Some genius designed this rivet so that the upsetting process begins at the blind side work face rather than at the end of the rivet. The result is that one length of rivet covers a wide range of work thicknesses. It is a relatively strong, efficient, good looking and convenient rivet. Purchased in lots of 1,000, it is also inexpensive (about three cents each for ⅛ in. diameter dome-headed rivets at the time of writing). I use nothing else for nonstructural applications. You will be amazed at how little time it takes to use up 1,000 rivets.

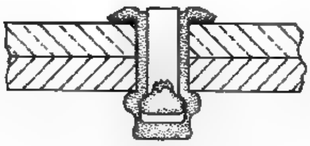
I use Avex dome head and large flange rivets. Their flush rivets are useless because they do not have enough metal in the head and invariably fail early. Avdell's patent must be in public domain



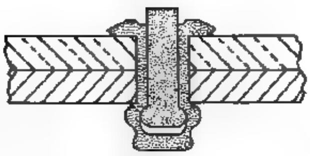
The basic blind rivet. Pictured is Cherry Commercial Fasteners' Cherry nail rivet. Cherry



Open-End Rivets

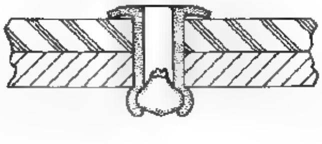


**Closed-End Rivets
Hollow-core**

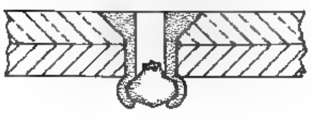


**Closed-End Rivets
Solid-core**

Dome Head

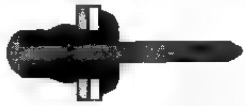


Countersunk Head

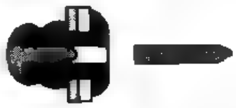


HOW IT WORKS

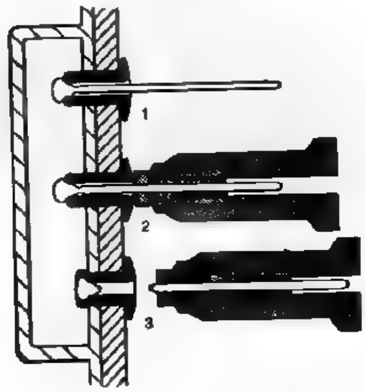
1. An Avex rivet is inserted in a prepared hole and the placing tool nose is slipped over the stem. When actuated, the pulling jaws grip the stem drawing it into the tool.



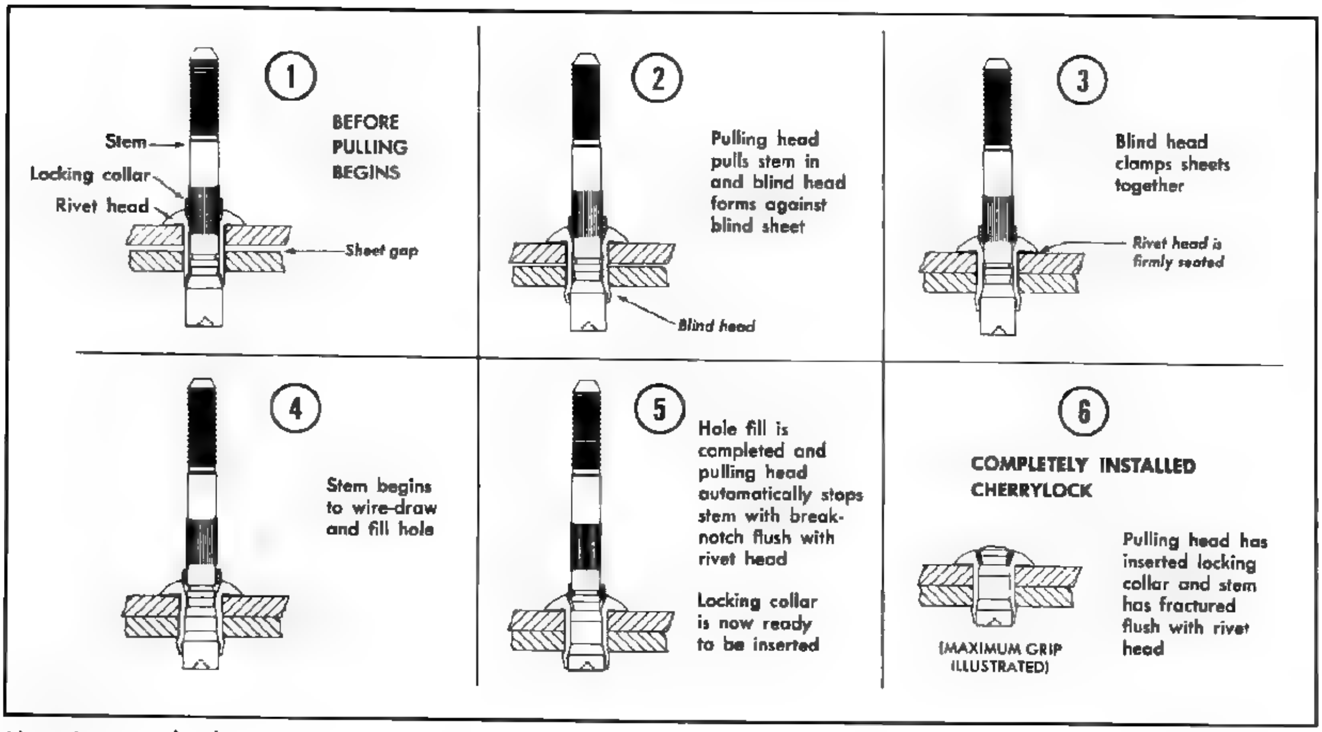
2. The stem *compresses* the shell, initially expanding it into 360° contact with the parent material. When the hole is completely filled, a symmetrical blind side tail forms and a *clamping force* is exerted from both sides of the work.



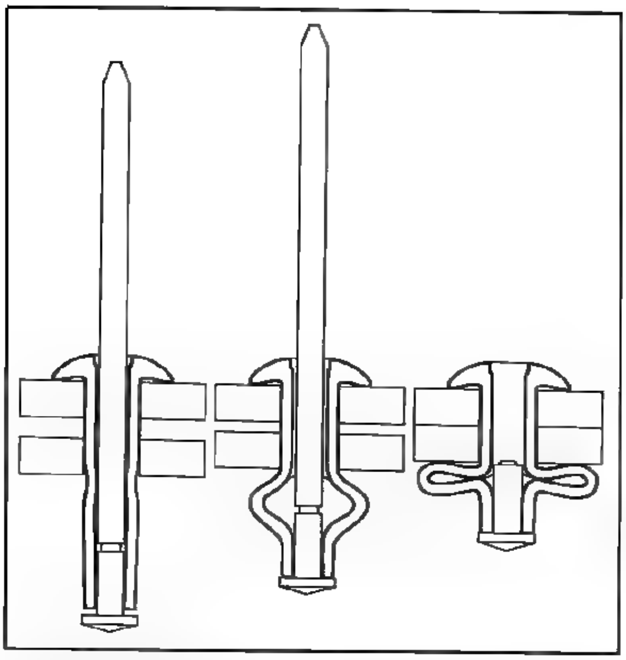
3. Finally, the stem separates, and its head remains permanently locked inside of the shell, sealing the bore. Installation is then complete, and optimum clamp up and vibration resistance is achieved.



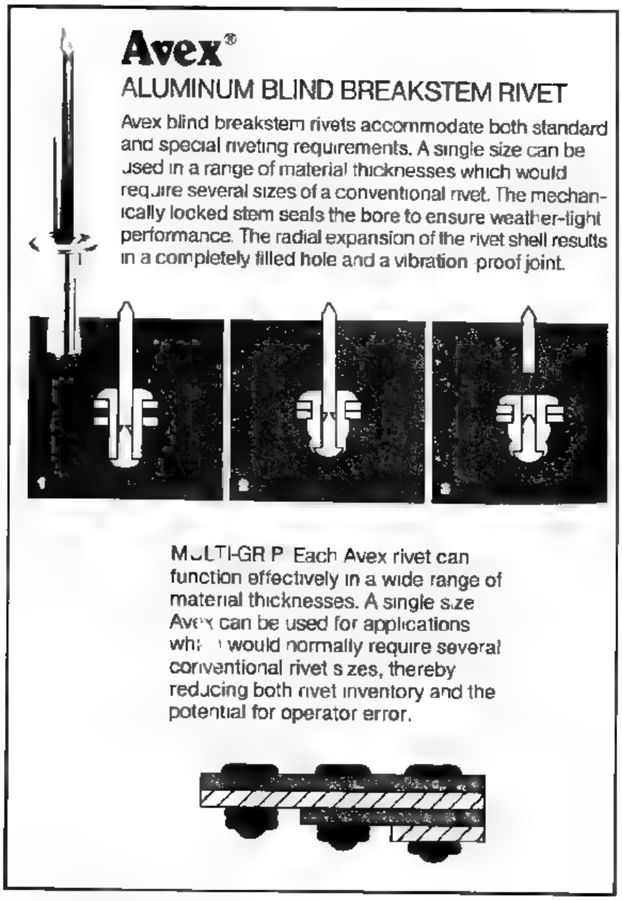
The basics of the blind rivet. Cherry



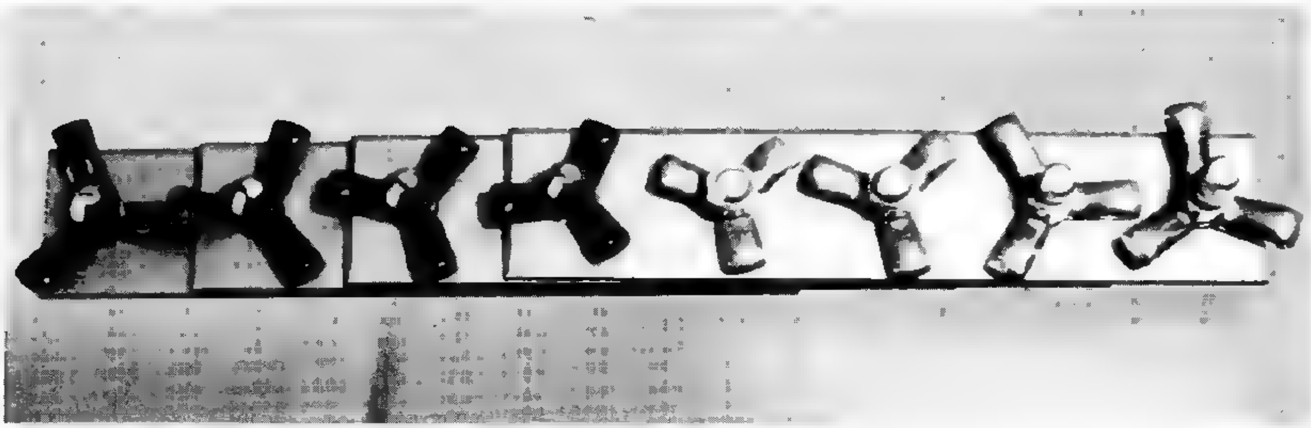
How rivets work. Cherry



The Cherry Klamp-tite rivet (non-structural version). Cherry

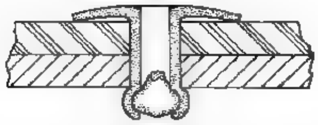


The Avex rivet by Avdell.



Riveting to soft materials.

Large Flange Head



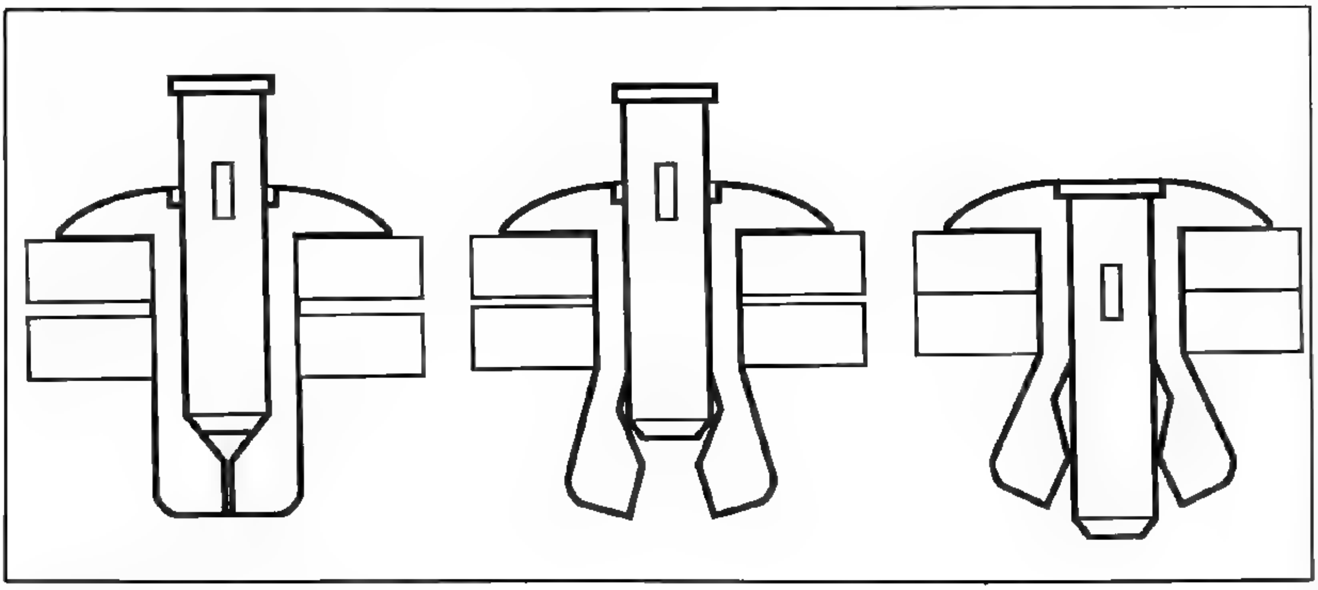
Large Flange POP Rivets have twice the under-head bearing surface of comparable dome-head rivets and are designed for applications where soft or brittle materials must be assembled to a rigid backing material.

Large-flange-head rivet for soft materials.

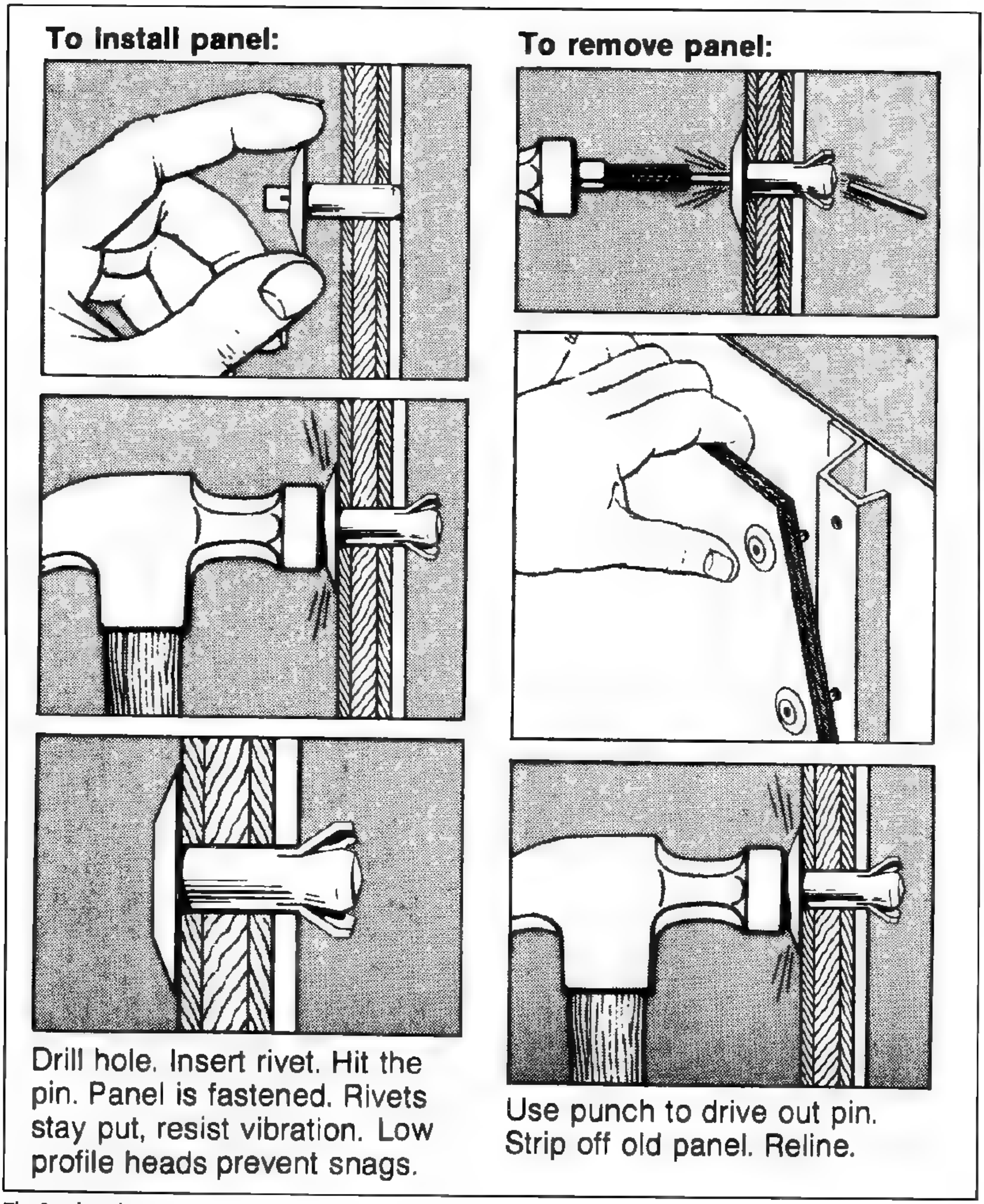
now, because other manufacturers are offering the same rivet. Premier buys them from Avdell and calls them multi-grip rivets, but the price is steep.

When riveting to fiberglass, use either a large area back-up washer between the rivet head and the glass or use a large flange rivet head. In any case, a back-up washer *must* be used under the shop-formed head with fiberglass (or any soft material) because a standard shop-formed head will pull through the glass almost instantly. Avdell's trick Bulbex rivet forms an oversized shop-formed head and does away with the necessity for back-up washers. If you are going to be doing a lot of riveting to soft materials, this rivet is probably your answer.

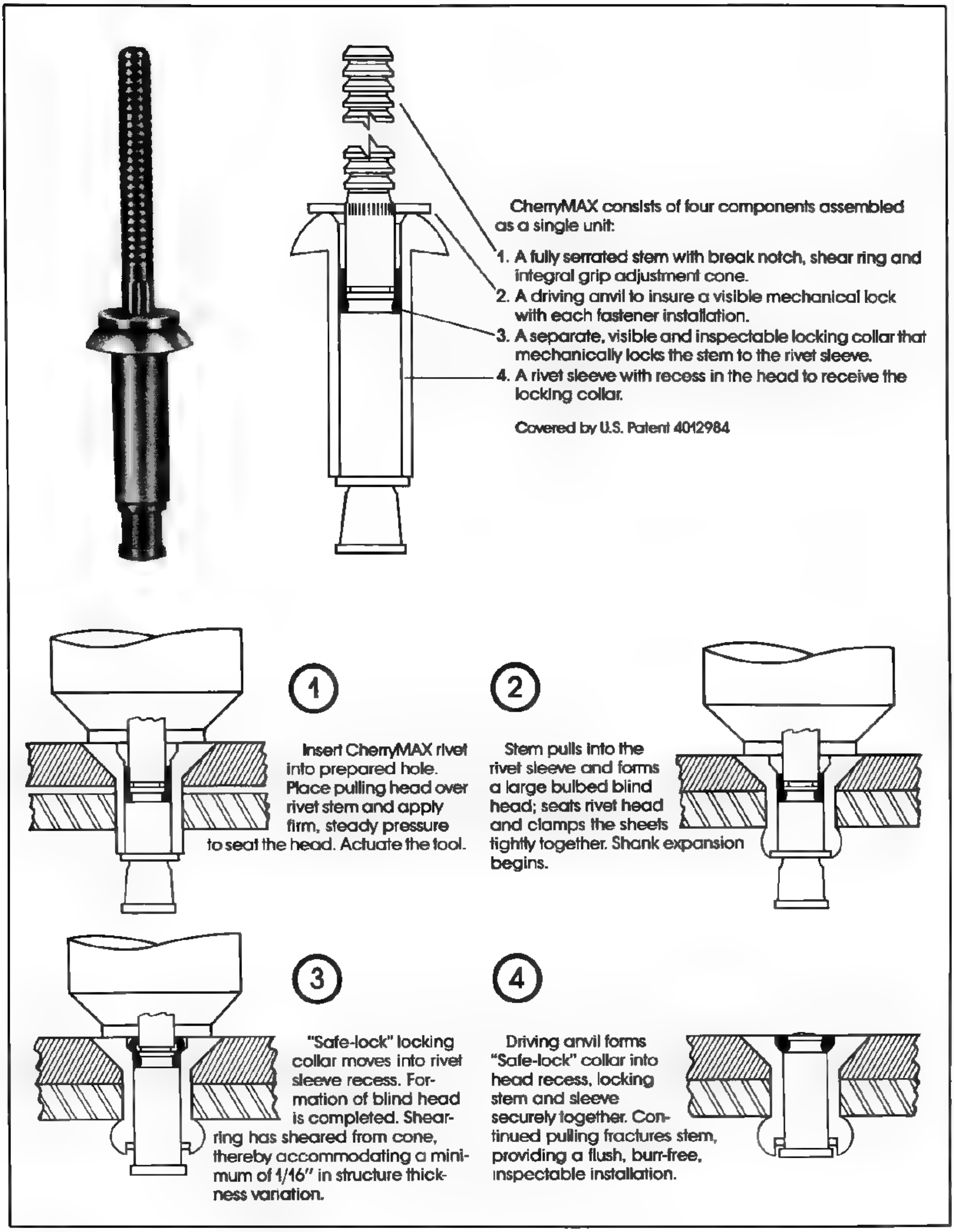
The trouble with the nonstructural blind rivet is that you get just about what you pay for—not much. The mandrel is not particularly well retained and always breaks well outside the shear plane.



The Cherry drive rivet. Cherry



The Southco drive rivet.



The Cherry Max aerospace structural rivet. Cherry

When first installed, some of them claim pretty impressive shear strength numbers (which I view with some suspicion). None of them are impressive when it comes to resisting fatigue. They are fine for scoops, spoiler, trim and such but are just not suitable for structure.

Since the mandrel is not particularly well retained in the shop-formed head of the rivet, you must also be aware that vibration can cause it to loosen and fall out. Since the mandrel was not in the shear plane to begin with, this will not have much effect on the strength of the rivet. However, if the rivet happens to be in the wrong place—like the carburetor air intake or plenum—the departed mandrel can lead to expensive damage. If the loss of the mandrel remainder can have awkward results, punch it out after you have pulled the rivet—since it is not in the shear plane to start with, it won't hurt the rivet.

If you find yourself in the position of needing to perform structural repairs, and hardware store rivets are all that is available, use the Monel rivets with steel mandrels. They are a lot harder to drill out than their aluminum cousins but are about a third stronger in shear. Many of the homebuilt aircraft people (who do not ever expect to have to drill out a rivet) use Monel pop rivets exclusively. If you can't find the Monel rivets at the hardware store, try a marine supply house.

Drive rivet

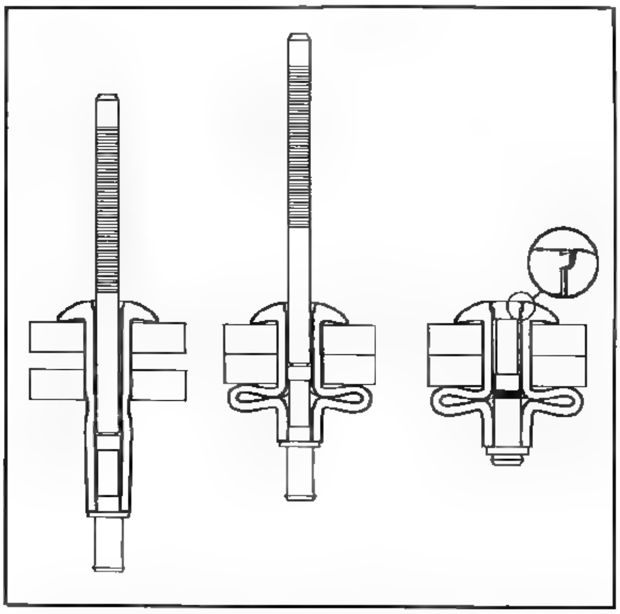
Most siding is attached to truck and trailer frames with Southco drive rivets. They are the easiest of all rivets to install and to remove, and are satisfactory for many industrial applications. They are not at all satisfactory for use on racing cars, however, or anything else where a failure can hurt someone.

Structural blind rivet

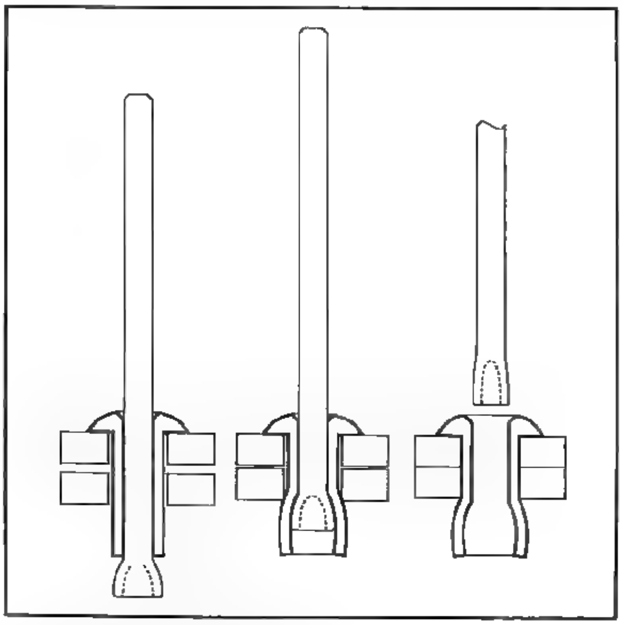
In the good old days of DC-3s and P-40s, all airplanes were put together with AD rivets. Both the engineers and the fabricators in the bird works considered all blind rivets to be structurally worthless. At the time, they were correct. Times have changed, however. We can now purchase a variety of MS blind rivets and one commercial rivet that are actually stronger than a solid aluminum rivet of equal size.

People have not changed, however, nor have their opinions. A large percentage of the race car designers and fabricators—most of whom learned from the old-timers—still believe that the only *real* rivet is a bucked rivet. Not me! I don't like bucking rivets. For one thing, I don't like the noise—it interferes with the music that I like to listen to while working. For another, I don't like the vibration. Lastly, I don't like drilling them out. I do, however, appreciate the fact that even when we consider the price of labor these days, the installed cost of a bucked rivet is still liable to be less than that of an aerospace blind rivet of equal strength. My attitude is that since I have to use a whole slew of structural rivets no matter what, I might just as well simplify my life by using the blind ones exclusively. Besides, once I have made that decision, I can sell my bucking equipment and have fewer boxes to carry around.

The leaders of the aerospace-quality blind rivet industry are the Huck Manufacturing Company and the Cherry Rivet Division of the Townsend Corporation. Cherry rivet has become the generic term for aerospace-quality structural blind rivets. The original aerospace blind rivet was the



The Cherry Klamp-tite rivet (structural version). Cherry



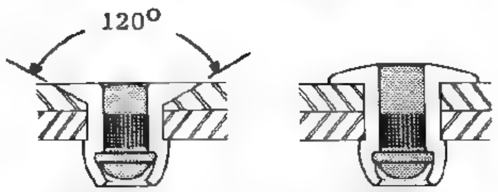
The Cherry C pull-through rivet. Cherry

CHERRY Q RIVETS
Structural — Self-plugging rivets

HIGH SHEAR STRENGTH
Cherry Q Rivet mandrel plugs the entire length of the rivet sleeve, providing full shear strength values for structural or load-bearing applications.

SEALING CAPABILITY
Specially designed mandrel of the Q Rivet is engineered to effect a seal, upon installation, which offers resistance to leakage.

VIBRATION RESISTANT
Rivet sleeve curls over end of mandrel to insure its positive retention.



FLUSH HEAD **PROTRUDING HEAD**

CHERRY Q RIVET IDENTIFICATION CODE

First letter is rivet material:
A = 5052 Aluminum B 5056 Aluminum C – Stainless
M = Monel S = Steel
Second letter is mandrel material:
A – 7178 Aluminum S – Steel C = Stainless
Third letter is head style:
P – Protruding L = Large C – Flush
Fourth letter is type of rivet:
Q = Cherry Q Rivet, structural, self-plugging
First number is rivet diameter in 32nds of an inch:
For example, –6 is 6/32nds or 3/16" diameter
Second number is rivet maximum grip length in 16th of an inch

For example, –8 is 8/16ths or 1/2" grip length

MINIMUM RIVET SHEAR AND TENSILE STRENGTH (lbs.)
CHERRY Q RIVETS

RIVET DIAM.	BS SERIES ALUM RIVET STEEL MAND. GRADE 19		MS SERIES MONEL RIVET STEEL MAND. GRADE 40		CC SERIES STAINLESS RIVET STAINLESS MAND. GRADE 51	
	Shear	Tens.	Shear	Tens.	Shear	Tens.
1/8	350	325	650	525	700	600
5/32	525	450	950	900	1050	1000
3/16	750	650	1450	1100	1650	1300
1/4	1250	1050	2350	2150	2450	2250

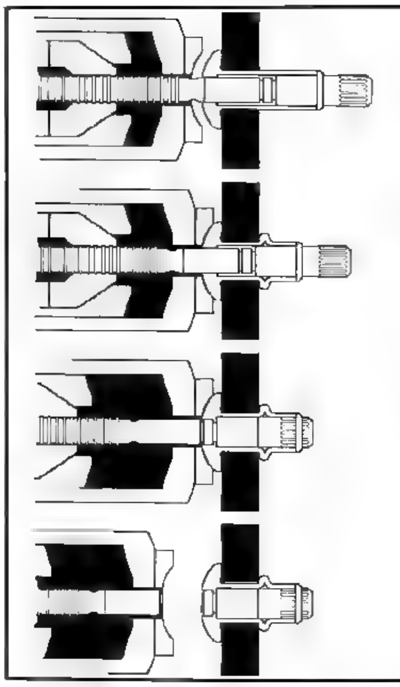
The Cherry Q structural self-plugging rivet. Cherry

Cherrylock. They also manufacture the Cherry Max, which is also MS and NAS certified. The Huck Corporation equivalents are the Huck-Clinch rivet and the Unimatic blind rivet. All of these rivets and their aerospace cousins are manufactured and inspected to exacting government aerospace specifications, and they are brutally expensive, starting at about forty-five cents each in lots of 1,000 and going skyward from there. The forty-five-cent ones are also hard to find and the manufacturers are little help. As you might expect, these rivets require special (and expensive) pulling tools.

Avdell supposedly manufactures an aerospace-quality lock-stem structural blind rivet called the MBC series. I have their data sheets, which is more than their Los Angeles area distributors can say. Unlike their nonstructural Avex cousins, they don't seem to offer any advantage in grip length tolerance, and I have been unable to obtain price or delivery information. Fortunately, for our purposes there are blind rivets available at more nominal cost than the aerospace items that will do the job for us just as well.

Commercial structural blind rivet

The major difference between a blind rivet that I consider to be suitable for structural use and one that I do *not* is that the mandrel or stem of the structural rivet will be securely held in place and will extend through the shear plane of the work. In other words, for a rivet to be suitable for structural use, the pulling mandrel must break off above the top of the work surface and must be securely retained in the rivet.



The Huck Magna-bulb industrial rivet. Huck

My current favorite nonaerospace structural rivet is Cherry's commercial Cherry Q rivet. The rivet body is available in 5056 aluminum, plated low-carbon steel, Monel and stainless. The aluminum rivets are available with either aluminum or plated steel stems, while the steel rivets come with steel or stainless stems. The stem is knurled so that the rivet material is swaged into the stem as the rivet is pulled. This feature provides positive stem retention and, providing that the grip length is correct for the work thickness, the stem breaks flush with the top of the rivet head. It is available in both dome and 120 degree countersunk head styles in ⅛, 5/32 and 3/16 in. and in full range of grip lengths. In aluminum they are rated at 350 lb. shear for ⅛ in., 525 lb. for 5/32 in. and 750 lb. for 3/16 in. Cost is in the range of six cents each for ⅛ in. rivets in lots of 1,000.

Material for material, the Cherry Q has about seventy-five percent more shear strength and eighty percent more tensile strength than the best of the trim rivets. But, as I pointed out earlier, ultimate strength is only a part of the rivet picture—the solid core rivets come into their own when we begin to talk about fatigue. Anyway, I use both the BS and the MS series of Cherry Qs. Do not confuse the Cherry Q with the Cherry N, which is just another trim rivet.

Huck makes two truly excellent commercial lines of blind rivets called the Magna-lok and the Magna-bulb series. They are both multi-grip rivets, i.e., they are not grip length sensitive so that one length of rivet has a wide effective grip length. They both break the mandrel at or near the top of the shop-formed head so that the mandrel is retained in the shear plane. The differences are that the Magna-lok mandrel is positively retained by a lock ring while the Magna-bulb mandrel is retained by mechanical gripping of longitudinal sirations or splines on the mandrel—similar to the Cherry Q. The Magna-bulb has a uniquely large shop-formed head, which makes it particularly suitable for use with soft materials where it is not practical to use a back-up washer.

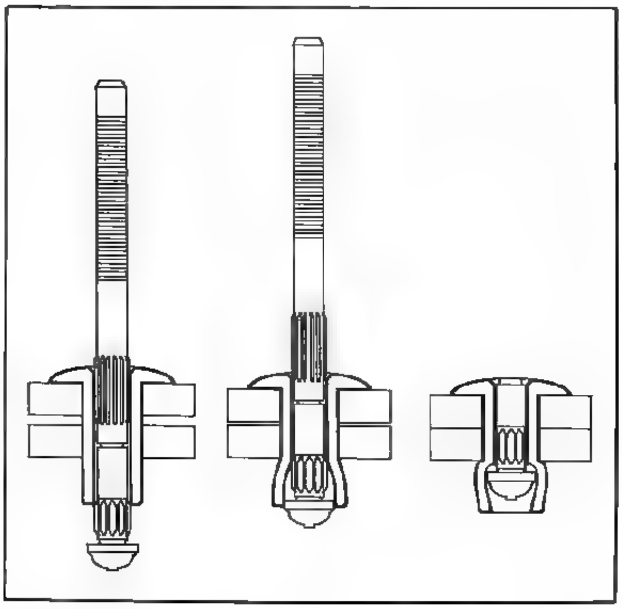
The only drawbacks are that the Magna-bulbs are available only as a steel rivet with a steel mandrel and only in ¼ in. diameter, which is a bit on the large size for most of us. Single shear strength under somewhat artificial test conditions is a rousing 3,600 lb. while tension is an equally impressive 1,900 lb.

The Magna-lok is available in a full range of materials and head configurations but only in 3/16 in. and ¼ in. diameters. In the 3/16 in. diameter, the aluminum rivet with aluminum pin has a shear strength of 700 lb. and an ultimate tension strength of 500 lb. These are outstanding rivets—if you can live with the diameters.

United Shoe Machinery, or USM, makes an excellent closed-end, solid core rivet line which is stocked by few, if any, of their distributors. This is a 5056 aluminum rivet with a carbon steel mandrel. It is available as dome head only in ⅛ and 3/16 in. diameters, and very few grip lengths. USM lists shear strengths of 480 and 890 lb., respectively. These are the highest claims that I know of outside of aerospace. The mandrel breaks well above the rivet head and must be cut off and trimmed after installation. This sounds like a big deal, but it is not. I trim with a pair of blacksmith's nippers that I have ground to a flush edge. I then grind off what is left of the mandrel with a hand-held grinder.

Cost is about four cents each for ⅛ in. rivets purchased in lots of 1,000, and they are very hard to find. Since the shop-formed head is closed, they are also a real bear to remove—the mandrel has to be driven through the shop-formed head before drilling commences. Unfortunately for this reason alone, I use these rivets as an emergency device only. USM also makes a closed-end hollow core rivet which looks exactly like the solid core item but offers only a fraction of the shear strength (tensile strength is the same). They make the hollow core rivet in both dome and 120 degree countersunk heads and in a variety of grip lengths. It will not do nearly the same quality job, however.

USM also makes an excellent structural rivet which they call the flush break. In this one, the stem breaks either at or just below the head of the rivet. The stem is well retained by the rivet, which swages itself around knurls on the stem (like the Cherry Q that it closely resembles). This rivet is really hard to find—and not worth looking for since the Cherry Q does the same job. Don't let the salesman confuse you by claiming that a break stem rivet is the same thing as a flush break. It isn't even close!

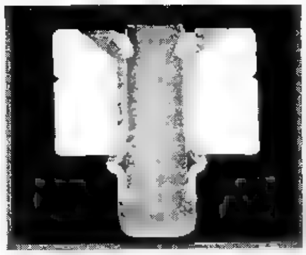


The Cherry Q rivet. Cherry

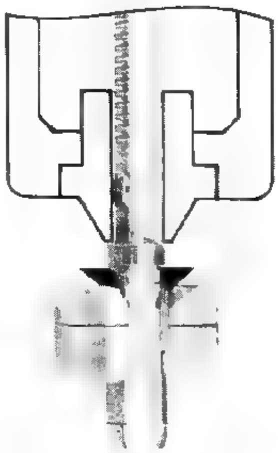
Positive-locking FOD free mechanical collar. The pre-assembled locking collar is mechanically locked into a pocket between the spindle and sleeve. This high-tensile, positive mechanical locking, non-frangible system is the reason you can use the Unimatic Blind Bolt with confidence in such FOD critical areas as jet engine inlets.

This cross-section photo of an installed Unimatic Blind Bolt shows the 'FOD free' collar seated to form a positive mechanical lock.

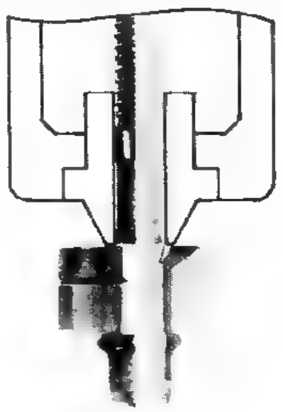
Tension-Tension Installation. The Unimatic Blind Bolt is installed with a *tension/tension* action, it does not rely upon a threaded spindle, sleeve and rotating tool to generate clamp force. It is not subject to loosening under vibratory conditions that occur in threaded torque-tension type fasteners.



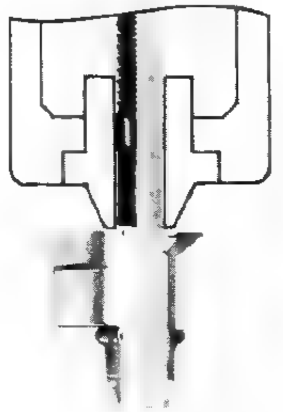
There are no threads on the spindle or sleeve of the Unimatic Blind Bolt to loosen under vibratory conditions. No threads to trap corrosive fluids.



As the installation tool begins to pull on the pin tail, the tool nose anvil bears completely on the locking collar. Continued pull causes the head of the spindle to upset the sleeve and form a strong bulbed head on the blind side.



As the blind side bulb is formed, the spindle is positioned so the nose assembly on the installation tool can drive the locking collar into the conical space between the recess in the sleeve and the locking groove in the spindle. This locks the parts firmly together.



The spindle is then separated in tension at the breakneck groove. There is no projecting spindle to be cut off in a separate operation. No frangible or FOD parts remain in the work place.

The Lockstem rivet.

Avdell distributes a structural rivet called, logically enough, the structural Avex. It combines the wide grip range of the standard Avex rivet with a flush breaking stem that is reasonably well retained in the set rivet. The only problem with this rivet is that they make it only in a low-carbon steel rivet with a medium-carbon steel mandrel. It is rated at 340 lb. ultimate shear strength in ⅛ in. diameter—which is not exactly earth shaking. They also market a structural rivet called the monobolt which is available in aluminum, but they make it only in 3⁄16 and ¼ in. diameters.

Lock-stem rivet

The strongest (and most expensive) aerospace blind rivets are of the lock-stem variety. This type utilizes a separate locking collar between the mandrel and the manufactured head to positively lock the mandrel to the head. They are fine for wet wings on hypersonic aircraft and such. Even if I could afford the things, though, they are not really suitable for use on racing cars as it is almost impossible to drive the mandrel out of the rivet after it is set; most of the rivet head has to be ground off first. Since driving the mandrel out is the first step in removing the rivet and since we remove a lot of rivets in the process of repairing damaged race cars, I, for one, do not want to know about lock-stem rivets in racing cars.

Aluminum-stem pop rivet

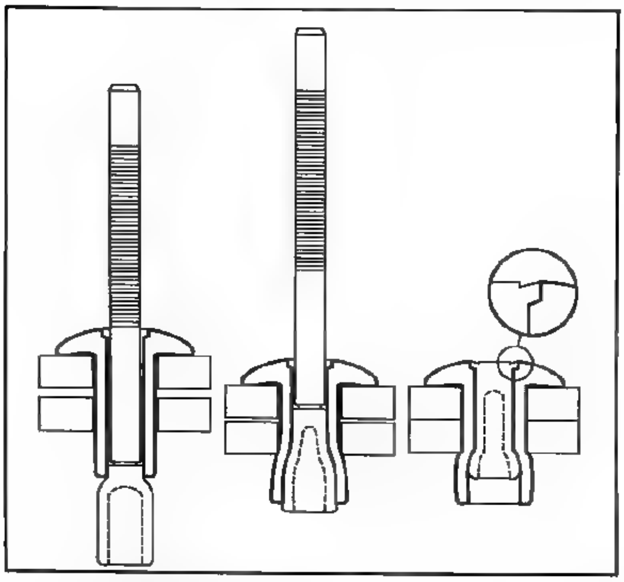
It is standard practice to use machine screws and plate nuts for inspection access holes and that sort of thing. If, for some unfathomable reason, you intend to drill out rivets frequently to allow access or inspection, you should use aluminum stems and aluminum rivets. I do not place rivets with the intention of drilling them out—*ever.* But in many

applications I place them with the full realization that I may well wind up drilling them out for the purpose of repair. For this reason I do not tend to use many steel or Monel rivets. I also want my riveted joints to remain strong in fatigue-prone environments, so I do not use commercial rivets with aluminum mandrels.

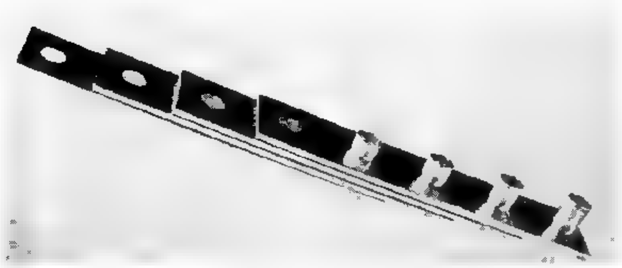
Countersunk or flat-headed rivets

There are many applications where the protruding rivet head is undesirable—sometimes for aerodynamic reasons, sometimes for space considerations and sometimes for the sake of appearance. There are pitfalls here. One of the things that we have to remember is that the aerospace industry uses a standard 100 degree countersink taper while the industrial standard is 120 degrees. Aerospace uses 120 degrees with composite materials. This means that neither the rivets nor the cutting tools are interchangeable.

We can seldom get away with cutting a countersink for flush rivets in the relatively light gauges of sheet metal that we are liable to be riveting. This means that we have to dimple the sheet metal in order to use flush rivets. You can purchase ⅛ in., 100 degree combination punch and dimpling die sets for your Whitney punch from Roper Whitney. The Eastwood Company sells a set of dimpling dies attached to a pair of vise grip pliers. As long as you can reach the spot to be dimpled with the short throat of the punch or pliers, these work well. Aircraft Spruce offers a neat 100 degree flush dimpling tool that fits in your hand pop-rivet pliers. You will also need several extra pulling mandrels. Theoretically, you should use a 120 degree countersink for the commercial rivets but, at least with aluminum commercials, the rivet material is malleable enough to conform to the 100 degree taper.



The Cherry Monobolt rivet. Cherry

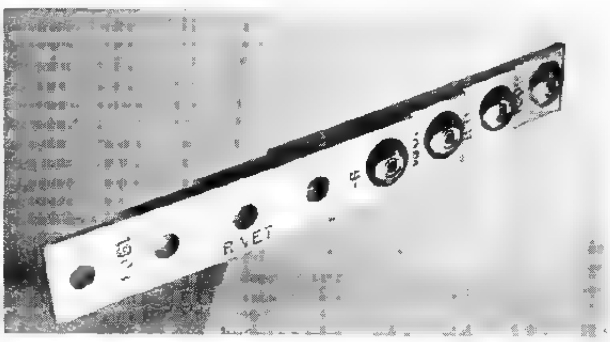


It has just dawned on me that it is not common knowledge that the cutting of countersinks in thin sheet metal is a bad practice. I had better explain. These drawings show the various ways of preparing sheet-metal surfaces for flush rivets. Example A shows the machining of a countersink in thin sheet metal which is to be joined to a similar sheet. There is precious little rivet bearing area left, and the sheets are actually being held together by blind faith. The same is true of example C, a thin sheet joined to a sheet of thicker cross section. If we reverse the situation, as in example C–1, everything is OK. Examples B and D show the right way to arrange countersinks in these cases. Section F illustrates that it is possible to dimple a thick sheet—but not with our hand equipment.

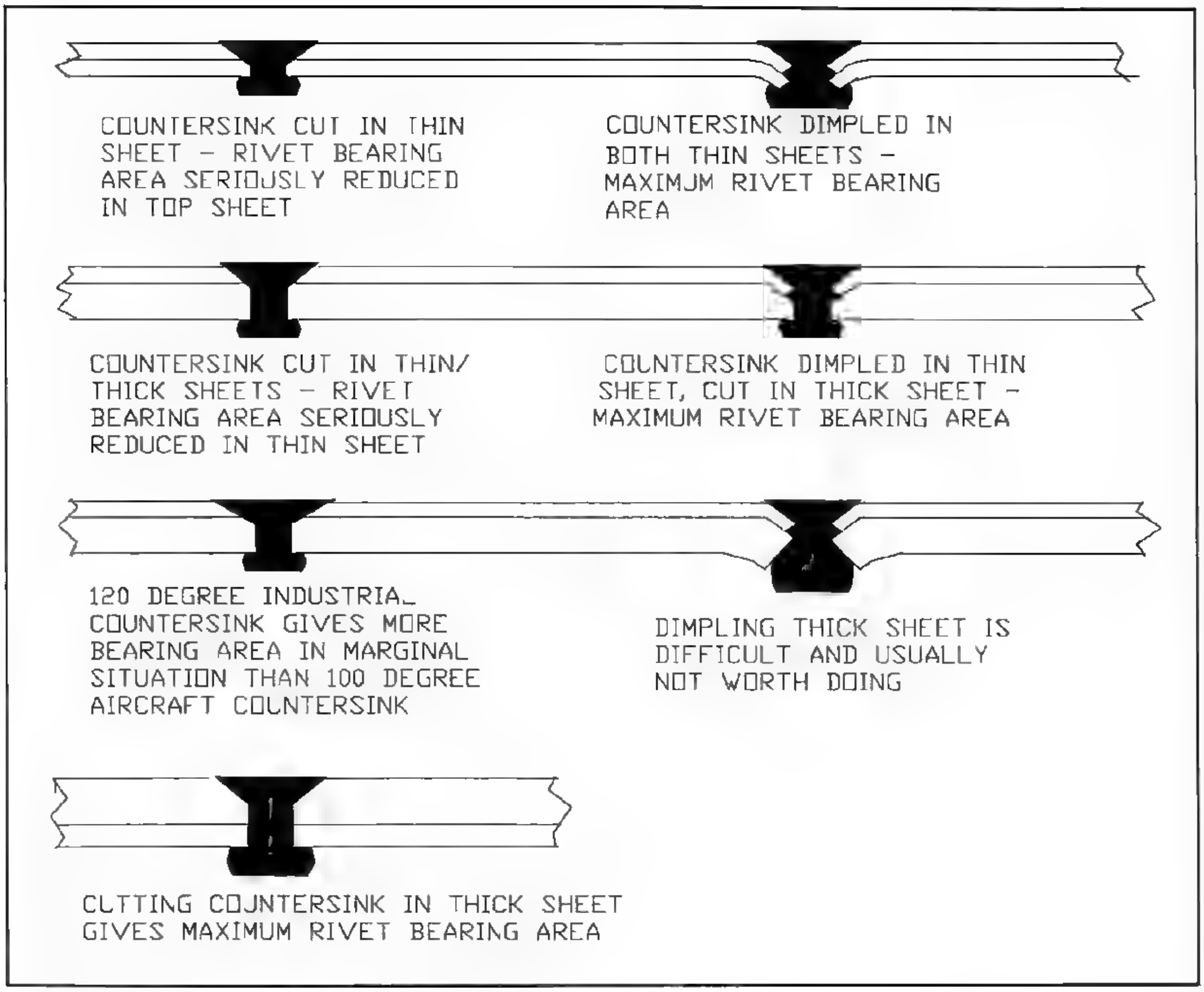
Dimpling is easy, although with 2024–T3, 6061–T6 and 7075–T6 you may have to locally anneal the dimpled area. As a point of interest, the aerospace industry uses an electric hot dimpler which locally anneals the area to be dimpled and then dimples it. Anyway, you cut countersinks in thin sheet metal at your peril. If you feel that you must do so, use commercial 120 degree rivets. They offer more bearing area than the aircraft spec 100 degree rivets and may save you from yourself—at least for a while. Cherry Qs are the best rivets available for this purpose.

Rivet holes

It may seem a bit basic to devote several hundred words to a subject as simple as drilling holes in sheet metal, but experience tells me that it is necessary.



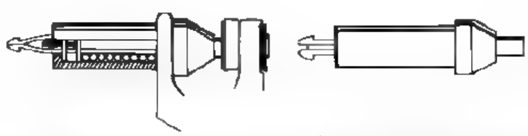
Other side of the structure, showing grip insensitivity.



Sheet preparation alternatives for flush riveting.

CLECO FASTENERS

Race cars and aircraft are mainly of riveted sheet metal construction. Cleco fasteners allow pre-assembly of riveted construction and trial fits. There is no way that you can fabricate structure from sheet metal and rivets without an assortment of Clecos.



Pushing the plunger extends the wires over the tee. In this position wires converge at the tip, making insertion or extraction an easy operation.

After insertion, release plunger. Automatic spring action retracts wires back over tee. Action of tee forces wires apart, locking fastener securely into hole, while pulling sheets firmly together.

Cleco skin pins.

The first rule of drilling satisfactory rivet holes is to use a sharp drill bit (#30 for ⅛ in. diameter rivets, #20 for 5/32 in. and #10 for 3/16 in.—except for Avex rivets which require #29, #21 and #8, respectively. It is normally best to predrill the holes in the smaller of the two pieces to be joined. Actually, it is better to punch the holes; you get a better hole and don't have to deburr it. The Whitney #5 hand punch is the riveters second best friend. Use even hole spacing laid out with a compass, a ruler, a pair of dividers or a template. An automatic center punch is an inexpensive big help, but buy a real one from a real tool store. The imports are junk. Anyway, if I am not in a big rush, I drill the first set of holes undersize (#40), transfer them to the other workpiece and drill the second set of holes, also undersize. Patience is the riveters *best* friend.

Clecos

The Cleco skin pin is a temporary rivet. Clecos are installed either with Cleco pliers or by a wing

nut/knurled wheel. They are used to hold panels in place both prior to and during riveting. Without them you cannot hope to do a decent job. They come in three variations, the edge Cleco, the standard plunger operated Cleco and the wing nut operated style. I use them all.

I drill or punch all of the holes (#40) in one of the panels. I then position the panel by hand, drill a couple of holes in the second panel and hold it in place with #40 Clecos. Next I proceed to drill the rest of the holes in the second panel—still #40 undersize—Cleco-ing the panels together as I go. When I have finished, I remove the Clecos, one at a time, and drill the final holes through both panels. A Cleco of the final size is then installed in the just-drilled hole and the procedure is repeated until completed.

It is best to start at the center of the job and work outward. This method ensures perfect hole alignment, and it is the only practical procedure that does. I remove the Clecos, deburr the holes, finish the edges of the parts, reinstall the Clecos and start riveting, also from the center outward.

Remember that you must deburr *every* hole and you must clean all of the swarf out from between the sheets of material. If the sheets are held apart by swarf or by burrs, you do not have a riveted joint, you have a failure looking for a place to happen.

Naturally, if I am in a hurry, as in pit lane repairs, most of this careful methodology goes by the board and I do what I have to do—as long as it is safe.

Rivet patterns

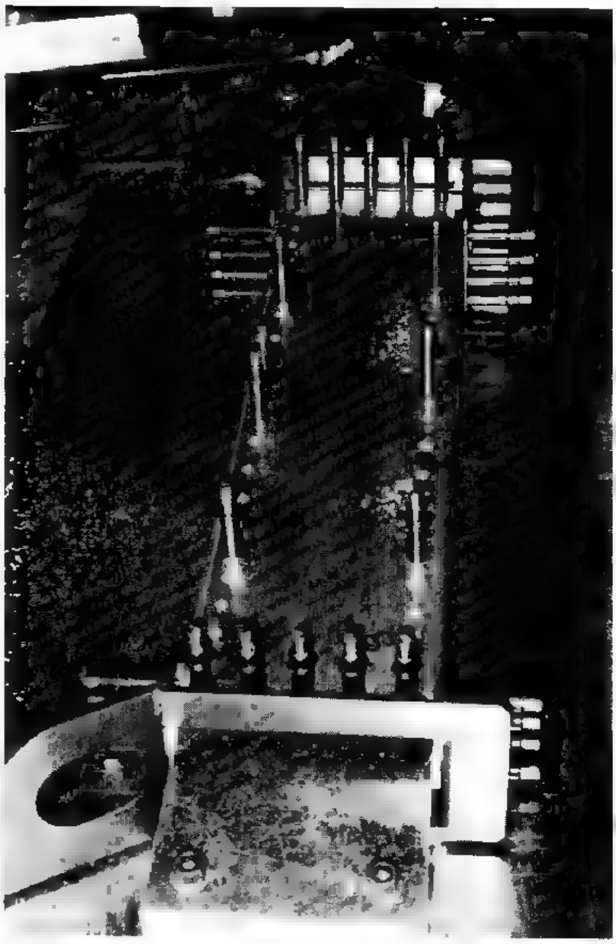
Rivets should always be installed in a regular pattern. Three factors are involved in the design of the pattern: rivet diameter, edge distance and pitch distance.

If you are going to realize the full strength of a riveted joint, you must select the rivet diameter as a function of work thickness. The rule of thumb is that the maximum rivet diameter should be from 2½ to 3 times the work thickness. The diameter, however, should never be less than the thickness of the thickest sheet involved.

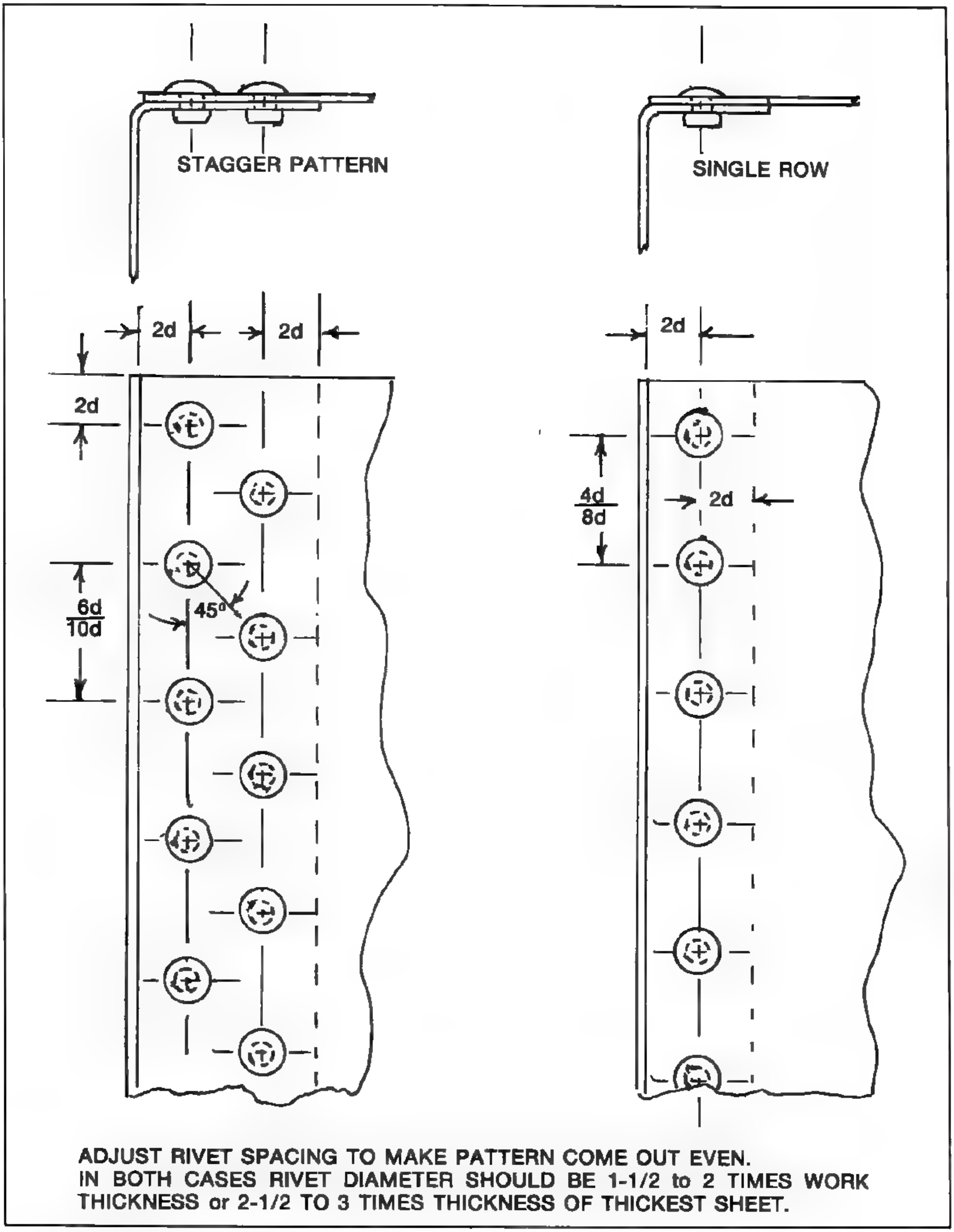
Edge distance is measured from the edge of the sheet to the centerline of the rivet holes, and it is critical. The rule of thumb in the aircraft industry is that the edge distance for protruding head rivets should be twice the diameter of the rivet shank. For flush rivets it should be 2½ times the rivet diameter. Insufficient edge distance will drastically reduce the strength of the riveted joint. Further, it may allow the sheet to distort or, in the case of hard rivets, even to crack due to the expansion force of the rivet and the hammering action of the gun. Excessive edge distance means excess weight and may allow the sheet to curl.

Pitch distance is the distance between the centerlines of adjacent rivets. The rule of thumb calls for a minimum pitch distance of three times the rivet diameter. Normal pitch distances are from eight to ten times the rivet diameter. The minimum distance between rivet rows is three rivet diameters, with four to six diameters being normal. Too great a pitch distance reduces the clamping force between the sheets of materials, which causes a corresponding reduction in the ever-critical friction between the joined sheets. The result is a weakening of the joint and possible fretting. In severe cases, insufficient pitch distance can permit buckling under load and the formation of gaps in the sheets between the rivets. Insufficient pitch distance can be just as bad, since there will not be sufficient metal available to properly distribute the load between rivets.

The diagrams shown here illustrate commonly accepted aircraft in laying out rivet patterns. Even though current aircraft theory states that there is no structural advantage to staggering rivets in adjacent rows, I still do it—if for no other reason than I find staggered rows aesthetically pleasing.



Installed Cleco skin pins.



Aircraft specification rivet pattern layout.

Rivet pattern templates

Laying out a precise pattern of rivets is a lot more work than laying out a close approximation by eyeballing it. I am willing to admit that the difference is not measurable in terms of performance, joint strength or fatigue life. The difference in appearance, however, is not only measurable, it is noticeable from a distance. Fortunately, rivet patterns are repetitive. Over the years I have made several rivet pattern templates from ⅛ in. flat steel cold-rolled bar in various widths and lengths. This means that I usually get to lay out each rivet pattern only once—which, to my mind, is the right number. Like everyone else in the world, toward the end of each rivet row I adjust my rivet patterns so that the end rivets come out where I want them.

Rivet insertion

I have recently taken to dipping each blind rivet in epoxy before I insert it into its hole. This makes sure that the hole is completely filled and seals the rivet against moisture. It also makes it damned near impossible to punch out the rivet when you want to. The airplane people dip their rivets in zinc chromate paste which seals but does not retain the rivet. OSHA and the EPA won't let us buy zinc chromate anymore so it is a moot point.

Tools

If you are shooting a lot of blind rivets, you may want to consider the purchase of either a pneumatic or a hydraulic blind rivet puller. Aircraft Spruce offers the Trojan Model HR-77 hand-operated hydraulic unit for about $200. It doesn't work very well, though. When it comes to pneumatic units, the one to avoid is Rodac (I have seen the pot metal castings explode under normal air line pressure). Mine is made by USM, is the bottom of their line, will pull 3/16 steel structural rivets and has given me twenty years of excellent service.

Premier Fastener Corporation, whose threaded fasteners I refuse to use, offers a hand-operated blind rivet squeezer that is superior to all of the others that I have tried. It is made in England, has a long snout so that it can reach into tight places and long handles so that normal human beings can pull even 3/16 in. steel mandrel rivets. It can be purchased, with a certain amount of coaxing, from your local Premier distributor.

Every so often I come up with the need to squeeze rather than to buck a solid rivet—like on the trailing edges of wings. I used to borrow an expensive air/hydraulic rivet squeezer from Jan Rury. Rury got smart and moved to Tahoe. I then discovered that a compound lever welding toggle clamp does a super job on 3/32 in. and ⅛ in. solid rivets—flush, dome or flat head in the A configuration. It will squeeze AD rivets but only in 3/32 in. diameter. My clamp is about 8 in. long, and says Knu-Vise #P-1800 on it. I bought it at Douglas Surplus a long time ago. Any compound lever parallel arm clamp will do though; merely polish the pedestals, adjust the width to the desired finished rivet thickness and have at it—one-handed.

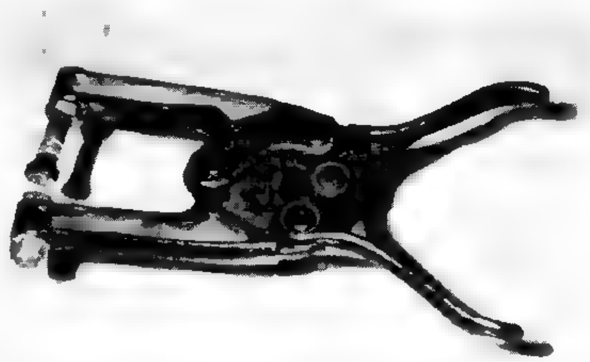
Rivet salesmen

Commercial rivet salesmen are about as unknowledgeable when it comes to their products as are commercial threaded fastener salesmen. Do not allow their hype to confuse you. If the rivet stem is not retained and locked in the shear plane, it is not a structural rivet. Further, don't let any of the snake oil salesmen confuse you with claims of tensile strength. The aircraft salesmen are, thank God, usually well informed and competent.

Strength of the rivet

For reasons that escape me, rivet strength is often expressed in both tension and shear. The strength of a rivet in tension is dependent only upon the strength of the manufactured head or the shop-formed head, whichever is stressed at the moment. No matter what the rivet material, only the cross-sectional depth of the head contributes to the strength of either the rivet or the riveted joint in tension. Rivets, like welds, are therefore weak in tension, and riveted joints in tension should be avoided. Tension applications are for bolts! The drawing also shows the easy way to convert a tension application to shear. We are not really interested in the strength of rivets in tension, though.

We are, however, vitally interested in rivet strength in shear. The theoretical shear strength of an installed rivet is determined by the diameter of the rivet, the strength of the rivet material and (if a mandrel is used and it is in the shear plane) the shear strength of the mandrel. The actual shear strength of an installed rivet is also dependent upon the fit of the rivet in its hole, whether or not the mandrel (assuming that there is one) is in the shear plane, how tightly the mandrel is gripped by the rivet material and the alignment of the rivet holes in the different layers of joined material. Some aspects of the picture are very simple indeed; for example, a hollow rivet with no stem is nowhere near as strong in shear as the same rivet with a closely gripped solid mandrel.



Homemade rivet squeezer.

Shear and tensile strengths of commercial blind rivets

Rivet description		3/32 in.	1/8 in.	5/32 in.	3/16 in.	1/4 in.
Cherry C pull through rivet,	Shear (lb.)	85	155	225	300	550
alum rivet, alum mandrel	Tension (lb.)	135	235	350	500	850
Avex dome head rivet, alum	Shear (lb.)					340
rivet, steel mandrel	Tension (lb.)					505
Cherry nail rivet, alum rivet,	Shear (lb.)	75	140	210	300	550
alum mandrel (pop similar)	Tension (lb.)	100	230	325	470	885
Cherry N rivet, alum rivet,	Shear (lb.)	100	195	300	425	825
steel mandrel (pop similar)	Tension (lb.)	165	290	430	720	1,100
Cherry N rivet, steel rivet,	Shear (lb.)	140	280	425	610	1,050
steel mandrel (pop similar)	Tension (lb.)	210	420	560	940	1,470
Cherry N rivet, Monel rivet,	Shear (lb.)	200	370	560	850	1,400
steel mandrel (pop similar)	Tension (lb.)	280	500	820	1,200	2,080
Cherry N rivet, stainless	Shear (lb.)	230	450	650	960	1,700
rivet and mandrel (pop similar)	Tension (lb.)	320	600	880	1,350	2,280
Pop closed end rivet, hollow	Shear (lb.)		305	430	575	1,300
core, alum rivet, steel mandrel	Tension (lb.)		385	605	840	2,000
(This is the typical English						
stressed skin rivet—wrong!)						
Pop closed end rivet, solid	Shear (lb.)		480		890	
core, steel rivet, steel mandrel	Tension (lb.)		385		840	
Structural Avex rivet, steel	Shear (lb.)		340	440	810	
rivet, steel mandrel	Tension (lb.)		385	520	750	
Cherry Q rivet, alum rivet,	Shear (lb.)		225	325	500	850
alum mandrel	Tension (lb.)		250	325	450	750
Cherry Q rivet, alum rivet,	Shear (lb.)		350	525	750	1,250
steel mandrel	Tension (lb.)		325	450	650	1,050
Cherry Q rivet, steel rivet,	Shear (lb.)		500	700	1,050	1,750
steel mandrel	Tension (lb.)		400	550	825	1,450
Cherry Q rivet, stainless	Shear (lb.)		700	1,050	1,650	2,450
rivet, stainless or steel mandrel	Tension (lb.)		600	1,000	1,300	2,250
Cherry structural T rivet,	Shear (lb.)				800	1,400
alum rivet, alum mandrel	Tension (lb.)				550	1,000
Cherry structural T rivet,	Shear (lb.)				1,000	1,900
alum rivet, steel mandrel	Tension (lb.)				700	1,250
Cherry Monobolt rivet, alum rivet,	Shear (lb.)				675	1,350
alum mandrel (Avdell similar)	Tension (lb.)				600	1,000
Cherry Monobolt rivet, steel rivet,	Shear (lb.)				1,500	2,750
steel mandrel (Avdell similar)	Tension (lb.)				1,350	2,400
Cherry Monobolt rivet, stainless	Shear (lb.)				1,450	2,650
rivet and mandrel (Avdell similar)	Tension (lb.)				1,150	2,350

Rivet strength in tension and in shear.

A rivet stem that is not tightly gripped by the rivet, or a rivet that is a loose fit in its hole (same thing) will eventually allow relative motion between the riveted components to develop. The result will be rivet failure from fatigue. Again, a rivet mandrel that has broken between the work surfaces will lend only the shear strength of a hollow tube of relatively soft rivet material to the shear strength of the joint. As usual, however, all things are not so simple as they at first appear.

Strength of the riveted joint

The shear strength of a riveted joint, on the other hand, is dependent upon all of the above plus the mechanical design of the joint, the spacing of the rivets and the spacing between the layers of joined material. This last spacing should be zero, but, unless an adhesive is used it seldom is.

Fail-safe

Standard aircraft practice calls for placing rivets so that they will be loaded in shear. All riveted joints on aircraft are designed to ensure that any failure will be a rivet failure rather than a material failure. This is desirable because aircraft are vastly overdesigned—they are designed for fatigue life rather than for ultimate strength. Further, the chance of the failure of a few isolated rivets leading to catastrophe is nonexistent due to the fail-safe design of all critical aircraft components.

Fail-safe simply means that there is a back-up system should the primary system fail. In this case, they use a lot more rivets than they need so that any loose or missing rivets will be noticed in scheduled inspections before the thing falls apart. The bottom line in the aircraft industry is that isolated rivet failure is much easier to detect and to remedy than incipient failure of parent metal.

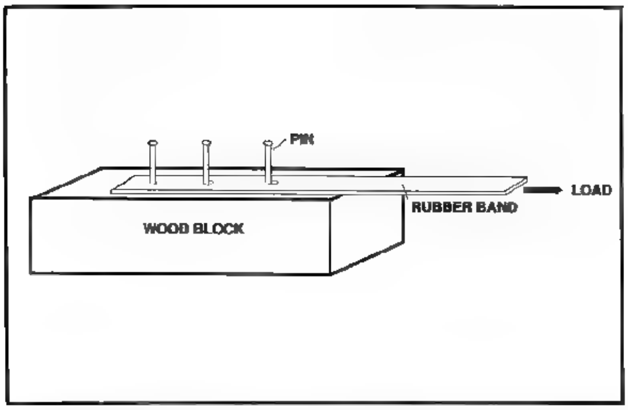
As usual, the design philosophy developed over eight decades of attempting to defy the law of gravity and extends to virtually all other structural fields. At one time I felt that there were exceptions to this rule when it came to race cars. I have changed my mind. Without exception, I want the first failure in any riveted structure that I design or am in any way associated with to be a rivet failure. I

RIVET DIAM.	USM 'POP' OPEN END ALUM RIVET ALUM MANDREL	AVEX DOME HEAD ALUM RIVET STEEL MANDREL	USM 'POP' OPEM END & CHERRY 'N' - ALUM RIVET, STEEL MANDREL	USM 'CLOSED END ALUM RIVET STEEL MANDREL	STRUCTURAL AVEX STEEL RIVET STEEL MANDREL	CHERRY 'N' OPEN END, STAINLESS RIVET, STAINLESS MANDREL	CHERRY 'Q' ALUM RIVET STEEL MANDREL	CHERRY 'Q' STAINLESS RIVET STAINLESS RIVET
3/32"	85 / 135		125 / 175			230 / 280		
1/8"	155 / 235		200 / 325	305 / 385	340 / 385	450 / 600	350 / 325	700 / 600
5/32"	225 / 350		325 / 450	430 / 605	440 / 520	750 / 1000	535 / 450	1050 / 1000
3/16"	315	340 / 505	430 / 650	575 / 840	810 / 750	1000 / 1300	750 / 650	1650 / 1300

Single shear and ultimate tensile strengths of commercial quality blind rivets. Strength is shown as shear/tensile. Note that the rivets which do not retain the mandrel in the shear plane exhibit greater tensile than shear t strength, and that this situation is reversed for those rivets in which the mandrel is retained and locked in the shear plane.

RIVET DIAM.	ALUM BUCK RIVET - AD UNIVERSAL HEAD	NAS 1900 SERIES ALUM BLIND RIVET	NAS 1900 SERIES MONEL BLIND RIVET	HUCK UNIMATIC BLIND RIVET ALUMINUM	HUCK UNIMATIC BLIND RIVET STEEL	HUCK UNIMATIC BLIND BOLT STEEL		
3/32"	217 / N/A							
1/8"	388 / N/A	495 / 325	1020 / 675	560 / 345	1050 / 675			
5/32"	596 / N/A	755 / 490	1565 / 1050	855 / 530	1665 / 1050	2340 / 1350		
3/16"	862 / N/A	1090 / 715	2260 / 1500	1165 / 715	2395 / 1500	3450 / 2100		

Single shear and ultimate tensile strengths of aircraft and aerospace buck and blind rivets.



Model illustrating uneven distribution of stress in multi-row rivet pattern. Cherry

will be damned certain that there are sufficient rivets involved so that the failure of any reasonable number of them between inspections will not lead to the rupture or separation of any part of the structure.

Design of riveted joints

You may recall that during our discussion of metal fatigue, I stated that I would cover fatigue and the riveted joint at a later time. That time has come. I have already stated that, although structural joins are never good, from the practical point of view they are inevitable. Let's see what is wrong with the riveted joint, and how we can use our newfound knowledge of metallurgy and strength of materials to minimize the unfortunate aspects.

The first problem with the riveted joint in sheet metal is that the loads involved are almost always tension/compression loads, which place the sheets in tension or compression but the rivets in shear. The tension stress in the sheet metal tends to elongate the round rivet holes into little ellipses whose major axes lie in the direction of load.

It is pretty obvious that single-row rivet joints are to be avoided. Not so obvious is the fact that, in the normal multiple-row riveted joint, the load is not evenly distributed among the rivets. This is true regardless of the spacing and/or pattern of the rivets, or of the number of rivet rows involved. The row of rivets nearest to the load is going to get a disproportionate share of the load and will therefore be more highly stressed.

This phenomenon can be graphically illustrated by my all-time favorite physical demonstration. If a heavy rubber band in the relaxed condition is pinned to a piece of wood by three equally spaced pins and then stretched, the three pin holes will be distorted to different degrees, with the hole nearest the load receiving the greatest amount of distortion. With the rivet pattern, while the first row will always receive an unfair portion of the load, by varying either the diameter or the spacing of the rivets, the level of stress can be more or less evened out among them.

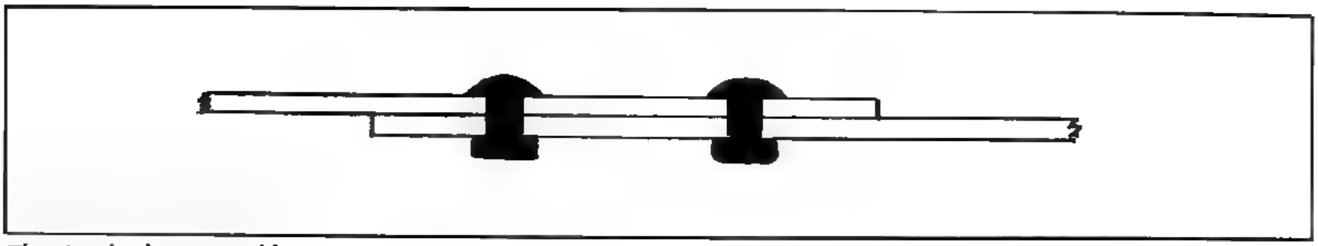
For our purposes, this solution is usually a lot more work than it is worth. Our usual problem with riveted primary structure does not have to do with varying stress levels in the rivets—or even with the strength of the rivets or of the joints. Rather, it is the problem of what happens to the rivets themselves in service. This problem is caused by inadequate joint design which is generally a product of cost cutting combined with ignorance.

In the construction of a racing car, the usual riveted joint is the simple lap joint—either a single row of rivets or an offset double row. It is quick, it is easy, it is cheap and it is inexcusable. The problem is not one of ultimate joint strength. Instead it is a down the road problem of what happens after the monocoque has a few thousand miles (and a few thumps) on it. The whole damned thing loosens up and becomes wanky or flexible is what happens—and it happens because, while the ultimate strength of the riveted joints may be perfectly adequate to withstand the loads involved, from the fatigue point of view the design is just plain not good enough.

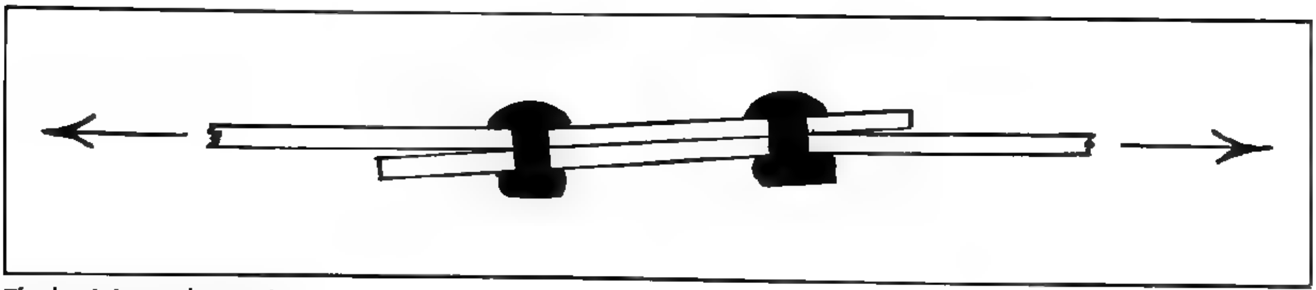
The inadequacy of the joint design is often compounded by the use of unsuitable rivets (like Avex rivets, which have no place in structure) and/or by unsuitable alloys of sheet metal. When a tension load is applied to the riveted lap joint, the two sheets attempt to align themselves with each other and, in so doing undergo elastic distortion, which creates a bending stress at the first line of rivets. This distortion is imperceptible to the human eye, does not approach the elastic limit of the sheet metal and, of itself, is harmless. If you tend to worry about it, don't *ever* look out the window at the wing of a commercial jet in flight.

The problem lies in the fact that the bending stress causes the load on the rivets to change from pure single shear to a combination of tension and single shear. Both the rivets and the rivet holes also undergo elastic distortion. This causes them to rub imperceptably against each other, and to wear. The more they wear, the greater the cyclic stress becomes until eventually, after many tens of thousands of cycles, each individual rivet is just a little bit loose.

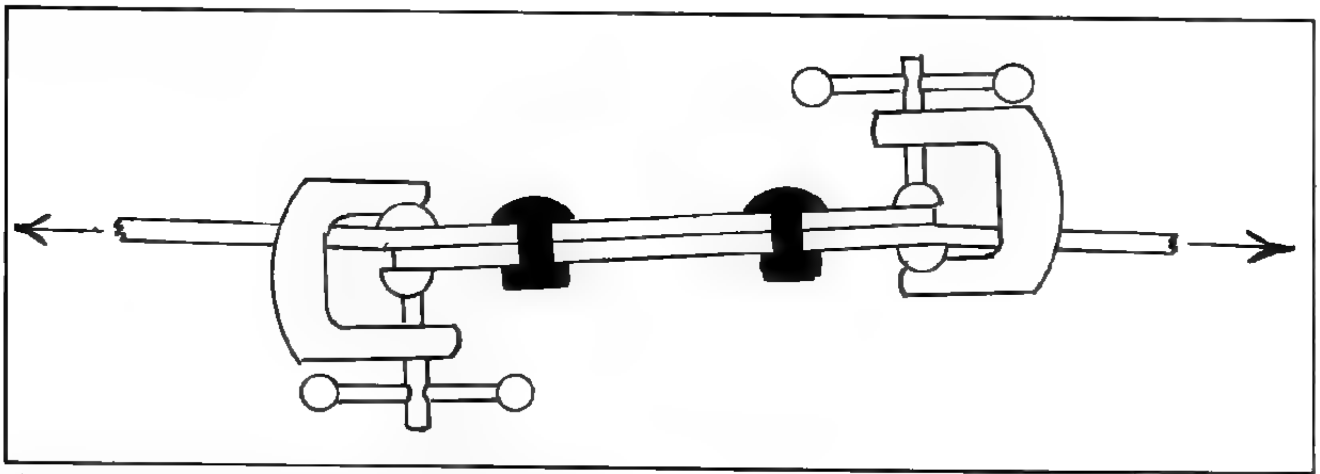
A rivet can be loose either axially so that clamping force is lost, or radially so that it is no longer a tight fit in its hole (or both). Little by little the whole structure loosens up and, if we happen to be talking about the monocoque, the whole car's responsiveness and general handling goes to hell in a basket. The usual syndrome is gradually increasing understeer over a long period of time. This is usually accompanied by a growing feeling of general sloppiness and an unwillingness to respond to changes in roll-couple distribution.



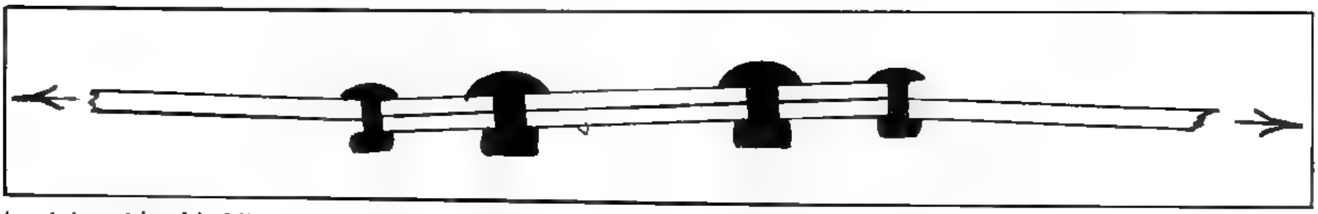
The simple sheet-metal lap joint at rest.



The lap joint under tension.



The lap joint under a tension load with the sheets clamped to prevent bending loads from reaching the rivets—the rivets are loaded in shear only.



Lap joint with added "stress confusing" rivets.

At the same time, any electrical ground that the designer was foolhardy enough to route through the main structure of the chassis is liable to deteriorate and become intermittent in operation, as will whatever circuit is dependent upon it, driving all concerned up the nearest wall.

The process is slow but steady and the improper use of adhesives to semibond the panels together only slightly retards the action (the proper use will help a lot). Once a tub has loosened, the only cure is to re-rivet the chassis—a tedious if not particularly difficult process.

A trick that works well on brand-new chassis is the insertion of a row of what is known in the aircraft trade as stress confuser rivets. If we could somehow clamp the edge of each sheet to the sheet beneath it then, while the sheets would still distort under tension loads, the distortion would take place in an area away from the rivets. This would separate the load carrying shear stress in the sheet metal from the distortion caused by bending stress, and subject the rivets only to shear stress—a better state of affairs altogether.

The use of a multitude of C clamps is impractical from the viewpoints of weight, cost and aerodynamics. The driving or squeezing of extra rows of rivets through the edges of the panels can accomplish the same end. The sheets will still bend imperceptibly under load, but, since there can be no such thing as shear stress through half a hole, the extra rivets can accept no loads other than the tension loads caused by the distortion of the sheets. The sheets will be held together at the edge of the lap and only the shear stresses will be passed along to the original row of rivets. These rivets are thus relieved of any tension load caused by bending stress and will function infinitely longer without loosening.

With a clean sheet of paper, one would not use the lap joint at all. A better way is the butt joint with doubler. For outside skins, this joint is also aerodynamically superior—so long as the doubler is on the inside. This joint is properly termed a single shear butt joint. While the sheet metal can still stretch (and the rivet holes elongate differentially) under a tension load and the shear stress is unevenly divided among the rivet rows, the sheets are aligned with each other and no bending stress will be developed under load, so the rivets will remain in shear.

The double shear butt joint is a stronger and more fatigue-resistant joint with more stable rivet support, but it has both aerodynamic and appearance disadvantages, and is heavy. Virtually the same results can be obtained by increasing the thickness of the doubler plate in the single shear lap joint. This method, however, still concentrates most of the load on the row of rivets set nearest the edge of the doubler—not good!

The best way to attack this problem is to use a tapered doubler plate and yet another row of rivets (of reduced diameter) near the edge of the tapered plate. Almost the same result is achieved by the use of two doublers when the first doubler is very thin in section. In actual practice, a single shear butt joint with the doubler plate twice the thickness of the sheets being joined works very well with a standard double-row staggered pattern of equal diameter rivets. It is less work and not much heavier.

Since most of our current racing regulations are written so that we end up with a certain amount of ballast anyway, I consider the couple of extra pounds that come along with the more fatigue-resistant rivet joints to be a more than a worthwhile tradeoff. I would feel the same way if the car ended up overweight by the amount of doubler plates and rivets involved—nothing can take the place of a rigid chassis.

I am not advocating the complete redesign of existing chassis. I am, however, advocating the addition of stress-confusing panel edge rivets when re-riveting a chassis. I also recommend giving a lot of thought to rivet quality and to joint design when re-skinning.

Hi-Shear rivet

At the begining of the hypersonic age, fastener engineers in the aerospace industry realized that the rivet as it was then known had reached the end of its rope. There were several reasons, chief among them being: first, aircraft skins were getting too thick to be satisfactorily riveted. Second, it was becoming increasingly difficult to arrange things so the fastener loads would always be in shear. And finally, skin temperatures were varying from the far subzero range of subsonic high-altitude flight to the friction-produced glow of hypersonic flight. This meant that the skins were going to change dimension a lot and that the fasteners needed to have the same coefficient of thermal expansion as the skins.

What was needed was a new family of fasteners that would combine the speed of installation and the permanence of a rivet with the shear strength of a bolt. The engineers also realized that fasteners with more installed strength in tension than offered by any of the then-current rivets would soon be necessary, and that fasteners would have to be made from the same materials as the structures they were fastening if they were to be capable of dealing with the wide extremes in temperature that the designers were beginning to visualize.

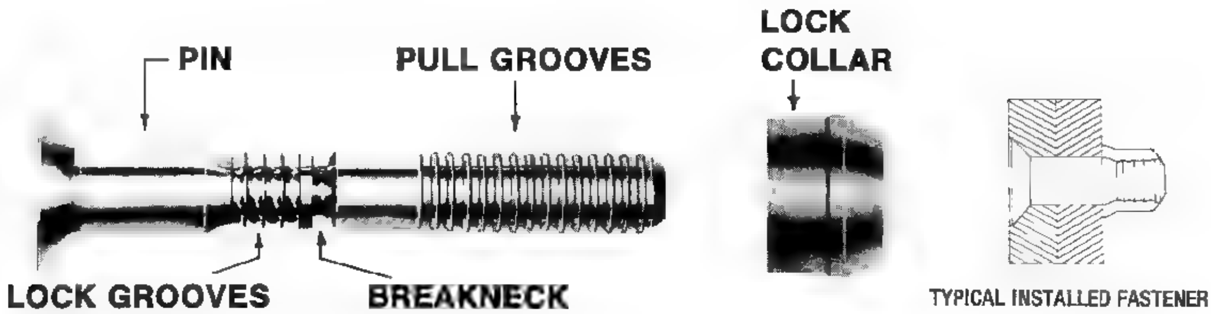
The answer was (and is) the swaged collar rivet. These devices are usually referred to as Hi-Shear rivets because they were originally developed and marketed by the Hi-Shear Rivet Tool Company. They are also currently manufactured by Cherry, Huck and Tridair.

The Hi-Shear rivet is made up of a pin and a collar. In use, the collar is swaged over the protruding end of the pin, which is furnished with a series of lock grooves into which the collar is swaged on assembly. The pin is not required to be malleable and so can be made from any material and temper desired. Hi-Shear rivets are installed like buck rivets, with similar tools (but a special bucking bar) and at the same speed. They are every bit as strong in shear as an aircraft bolt of the same material, heat treat and diameter. They are grip length critical and, like rivets, they are a permanent assembly. Like rivets, there is no torsional load or shear stress involved in installation, so the clamping load and the level of installed tensile stress are predictable, repeatable and independent of operator skill. The absence of shear stress means that there is no need to retighten the fastener after assembly.

There are many different types of these fasteners, and the pins and collars are available in every structural metal known to man. They are also designed for use in tension applications. Because the swage collar rivet is of no direct interest to the racer, I am going to end the discussion right here.

Rivet failure

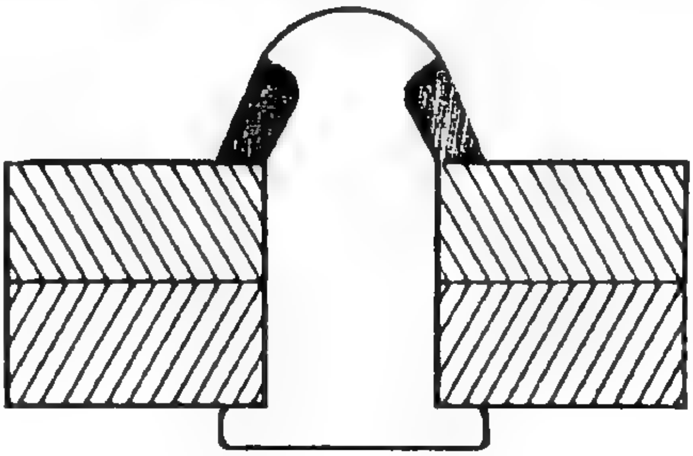
Most rivet failures are caused either by the use of the wrong rivets or by inadequate joint design. If the head comes off a rivet, it was installed in such a way that the rivet was loaded in tension, and you should have your wrist slapped. If the rivet fails in shear, you are not using a large enough (or a strong enough) rivet. If the rivet holes elongate, the sheet



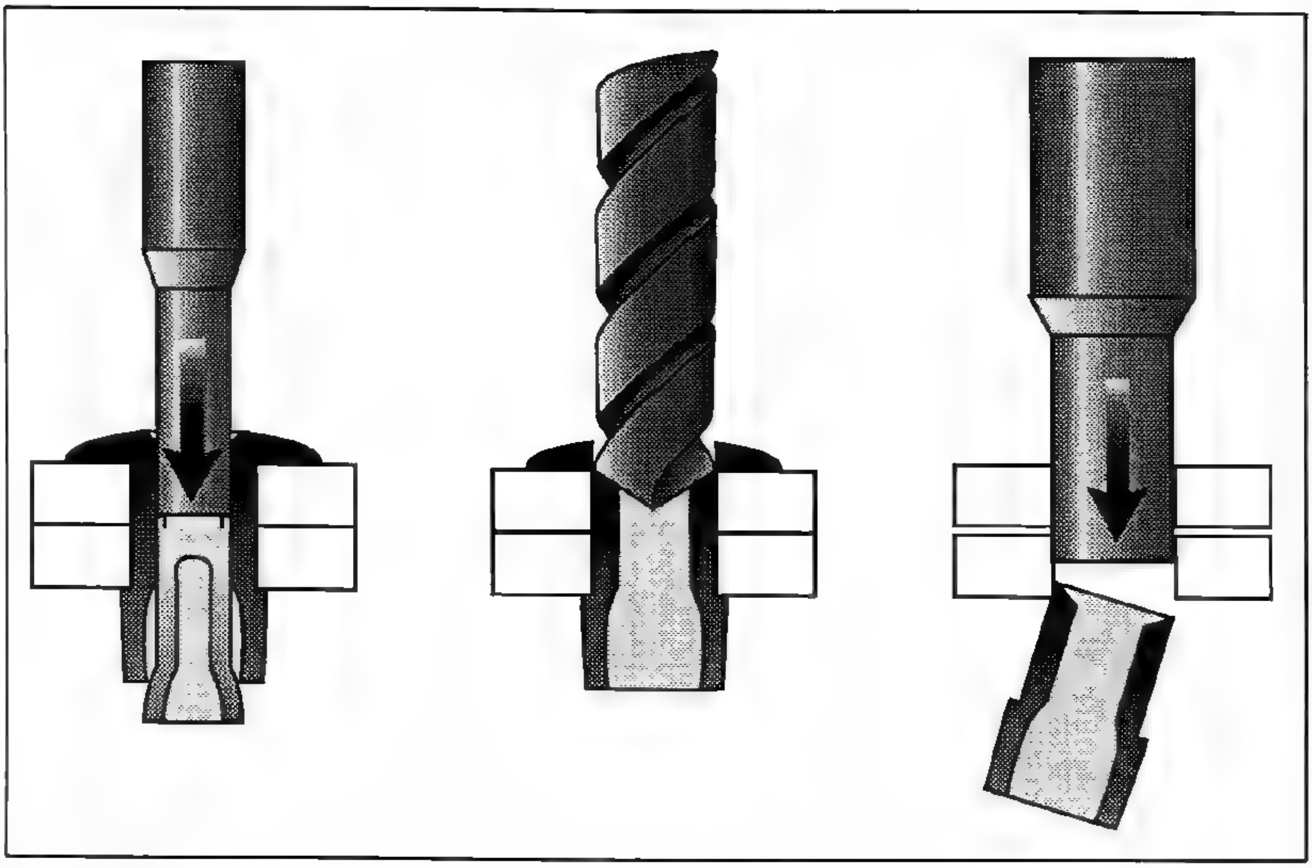
HI-SHEAR RIVET

The Hi-shear rivet is made of heat treated steel and heat-treated aluminum alloys, and as its name implies, is designed for high shear strength. The rivet is not up-set in the usual manner as applied to rivets, but instead is held in place by an aluminum alloy ring which is swaged into a locking groove at the end of the rivet. Fig. 17

FIG. 17.



The swage collar rivet.


Removing blind rivets. Cherry

metal has failed locally in bearing, and you need more bearing area—which means either a tighter pattern or larger rivets.

Usually the first sign of rivet or joint failure in shear or bearing is a telltale black ring of aluminum oxide around the rivet head. The oxide is a product of fretting and means that you had best re-rivet the assembly with larger rivets.

If you suspect rivet or joint deterioration but cannot detect any visual evidence, drill/punch out several rivets and have a look—one of the neat things about rivets is that they are easy to replace.

Rivet removal

Things being what they are, anything installed on a racing car will sooner or later have to be removed. As usual there is one right way and several wrong ways to do it. The procedure is different for solid and blind rivets but the object is the same: to neither enlarge the rivet hole nor mar the skin.

With solid rivets, I back the shop-formed head with a bucking bar, deepen the dimple on the rivet head with a center punch and drill through the rivet head with the same size drill that I used for the rivet hole. The head will fall off when the drill first goes through it and before it goes into the shank. I then back up the skin with a good hefty piece of bar stock into which I have machined a recess for the shop-formed head, and try to tap the rivet out with a punch. If it doesn't come out with a moderate amount of force, I drill through it with a smaller drill bit and then punch out the remains.

With blind rivets, any attempt to drill them out before you have removed the mandrel will result in dull drill bits, badly augered holes and, very possibly, the creation of some nasty little spiral marks on the skin from where the drill bit jumped off the mandrel. These are known in the trade as drill tracks and are a badge of shame. Drill tracks can also be caused by attempting to drill any hole without center punching first. Anyway, I back the sheet with something heavy and punch the mandrels out with a 1/16 in. straight punch. It is then simple enough to drill the head off. It is unlikely that it will punch out after the head is drilled off, but it is worth trying. If it won't punch out, I drill through with a 1/64 in. undersize bit to save the hole. If the people who drill out blind rivets without removing the mandrels are fools, then the people who use chisels to remove rivet heads are butchers.

If the holes do get augered in the process of rivet removal, the next larger rivet size must be used for replacements.

Instead of using a dolly (or nothing) to back up the panel when I am either punching the mandrel out of a blind rivet or knocking the stem of an already beheaded rivet out of its hole, I now use a couple of pieces of heavy bar stock with generously radiused corners and a few ¼ in. holes drilled in strategic locations. I hold the appropriate chunk behind the panel, wiggle it around until the shop-formed head of the rivet-to-be-removed falls into one of the holes, hold a punch on the other end of the rivet and hammer the thing out. Assuming that the back-up bar is properly positioned and firmly held, the mandrel or rivet pops right out. Using this system, it is almost impossible to distort the sheet metal.

When you are drilling off rivet heads, the drill bit ends up stacked full of rivet heads—which are a bear to get off. If you take the trouble to remove them after each operation you will find that, in groups of one, they come off very easily. I use a pair of diagonal cutting pliers to remove them.

Rivet nails

My last words on the subject of rivets have to do with flat tires! The most common single cause of flat tires on racing cars is the pop-rivet nails that someone did not pick up after pulling the rivets. Anytime that you are installing blind rivets, carry a coffee can or a dixie cup around with you and put each nail into it as the rivet is pulled. If you do not, I will guarantee that eventually a rivet nail will end up in a tire, a foot, a cheek or some equally undesirable location. If you wait until you have finished to pick up the nails, you will miss several of them—and Murphy will get you!

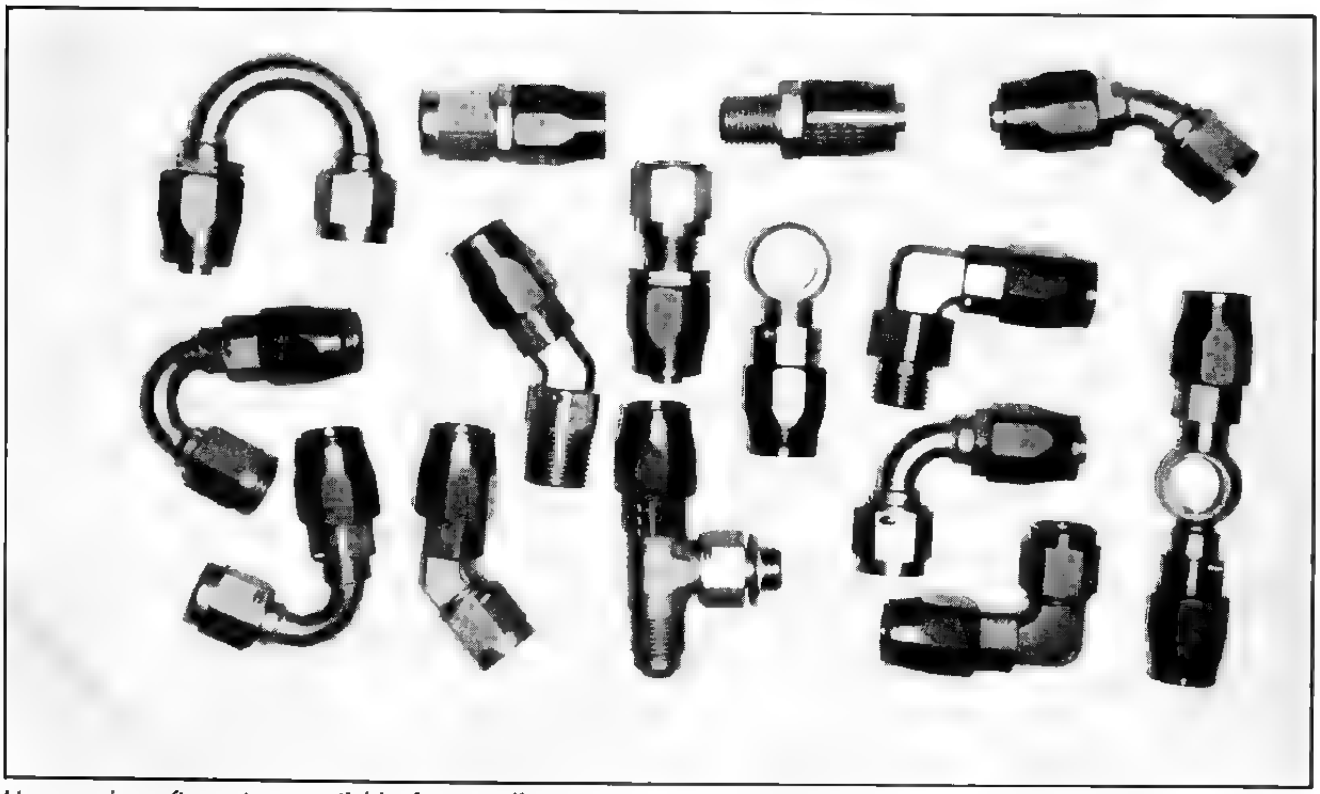
10

High-performance plumbing

Almost two decades ago I wrote, "If I were asked what single category of failure—other than brain—is the most common cause of race cars not finishing races, my answer would be 'plumbing failure.'" Just to prove that some things *do* change, if I were asked the same question today, my answer would be electronics. Whether this is because the racers have started using the right stuff in plumbing, or because we have a lot more electronics on the cars than we did then, I am not prepared to say.

In my world, the rather inelegant generic term plumbing includes all of the hoses, lines, tubes and fittings that transport fluids from one part of the race car to another. It doesn't much matter what the purpose of the machinery is; all that changes from a D-9 Caterpillar to a DC-9 Jetliner or from a Honda Civic to a Williams-Honda Formula One car are the volumes of the fluids, the rate and pressure at which they flow, and the temperatures involved. The object of the exercise is and always will be to get the fluid from where it is to where we want it to be with the minimum practical drop in pressure, while keeping all of it inside the system. The price of failure is the same in each case—the machinery stops.

For the purpose of this discussion, I have divided the family of plumbing lines, hoses, fittings and hardware into three convenient branches according to the type of fluid involved: coolant water, oil and fuel, and hydraulic fluid. I will confine our discussion to the general mechanics of plumbing; the design of systems is specialized and outside the scope of this book.



Hose end configurations available from Earl's Performance Products.

Coolant water systems

I know quite a bit about coolant water systems for internal combustion engines, and I assume that coolant water systems for anything else are similar. Water systems are about as simple as you can get. Maximum pressure involved is about 22 psi, and 14 to 16 psi is normal. Maximum temperature is about 256 degrees Fahrenheit—for a very short time. All that the lines or hoses do is carry hot water from the engine to the radiator and cooled water back to the engine.

With these requirements it should be impossible to go wrong. Not so. Water line failures are distressingly common and are almost always caused by use of the wrong materials, improper procedures or total lack of inspection. The rules are short and simple, but crucial: use only top-quality hose and hose clamps; and the ends of all metal tubes in the system must be beaded as shown, and as seen on any radiator inlet or outlet tube.

There are several ways to bead the end of a tube, ranging from the use of a commercial beading tool to laying a bead of weld around the tube. Radiator shops and most sheet-metal shops have beaders. If you do not bead the end of a tube, Murphy tells us that eventually the hose will blow off the tube. Murphy further assures us that this will happen either between three and four o'clock in the morning during a blizzard or while leading a motor race.

Water hoses

Until 1980, I said to use only Gates Green Stripe water hose on racing cars and to avoid molded-shaped hose like the plague. The reason was that the automotive hoses were simply not good enough for racing. That has changed. With the advent of the full smog engine, the underhood temperature(s) of street cars, domestic and imports alike, have risen to unprecedented and fearsome levels. The junky molded water hoses that the Detroit aftermarket loved just couldn't take this new environment at all. So the reputable manufacturers have developed a whole new generation of vastly improved water hoses—and bushings, elastomeric gaskets, fuel lines and so on. If anything good can come of our EPA emissions requirements, then this is it.

While I am still enamored of Gates Green Stripe water hose and use it exclusively for my straight-line applications, I now make extensive use of the formerly despised molded rubber hoses in various convenient shapes. But I use only those hoses with woven fabric cores and manufactured by either Gates or Dayco.

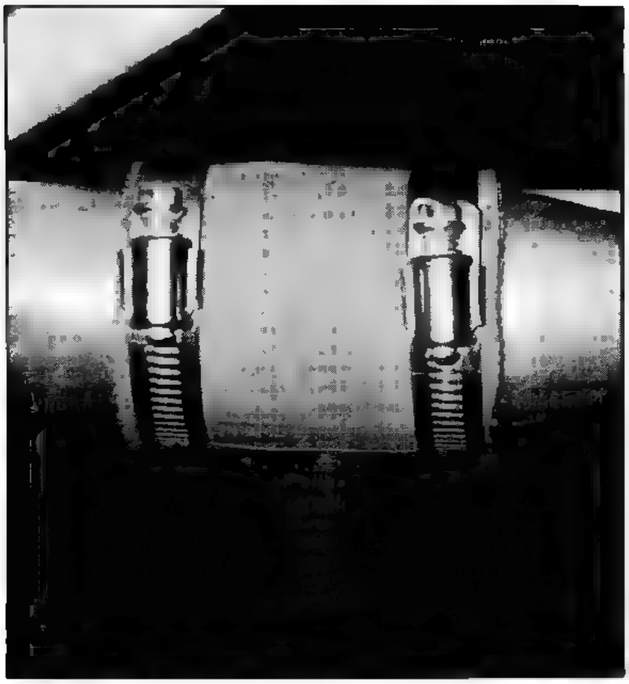
You are pretty safe using Detroit, Japanese or German original equipment water hoses, but you have to be a little bit careful here because dealers, being what they are, are liable to save a few bucks in the water hose department and stock a cheaper substitute rather than the OEM item. I do not use French, British, Italian, Taiwanese or South American water hose. The safe way is to buy from a good parts house—they will have a larger selection than anyone else anyway—and use nothing but Gates and/or Dayco. Pay no attention to what the "I've been selling this stuff for twenty-five years and I know what I'm talking about!" counter man has to say about either quality or availability.

I do not use the ever-popular corrugated or accordian-style "fits all" hose—except in emergencies. This is not because it doesn't work or even because it is inefficient. I don't use the stuff because it is just plain ugly. I do, however, carry a couple of lengths of it in the tow truck.

I use the exact, or at least a close approximation thereof, degree of bend that was molded into the hose. If the natural bend in a molded hose is either opened or closed very far, there is a real chance that the hose will collapse with entirely predictable results. This requirement to match angles means that we will spend an inordinate amount of time searching for the right water hose. The incredible lack of standardization throughout the automotive industry with regard to water hoses—and fan belts, oil filters, fuel filters and the rest of the automotive engine auxiliaries—ensures us that we will find it, eventually. The right hose is indeed worth the search. If I have any doubts at all, I insert a light coil spring inside the hose, especially if it happens to be on the suction side of the water pump. Any internal spring on the suction side of the water pump requires some sort of mechanical stop to positively prevent the spring from contacting the impeller.

I religiously replace even the best water hose at the end of each racing season, or every two years

Beading the end of a tube for hose retention.



Worm gear hose clamp—optimum construction but insufficient hose outboard of the clamps.

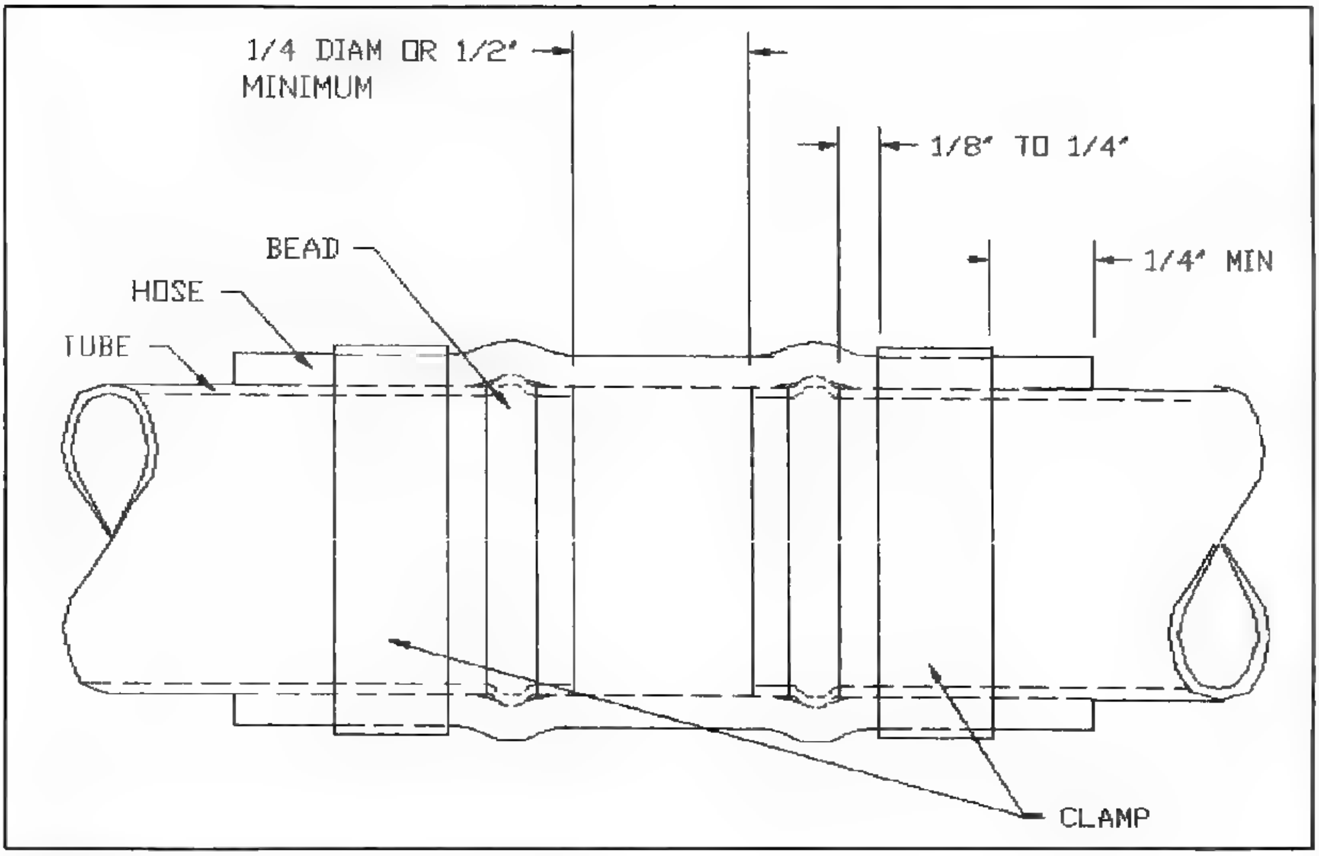
on my street cars, or when they start to feel the least bit mushy.

Pressure caps

Because I sincerely dislike burns on my body, I use the Stant brand of lever-releasing pressure caps. I use 14 to 16 psi caps. I don't use more system pressure because there is no way in God's green earth that you should need more. Besides, the 22 psi pressure caps are hard to find.

Hose clamps

I use only top-quality worm-gear-type hose clamps with through slots. The formed slots are a failure looking for a place to happen. Stainless steel is the only way to go. Ideal and Trident are good brands that are readily available. I prefer the hex-headed type that can be operated either with a screwdriver or with a 5/16 in. socket wrench. Make sure that the clamp is installed on the tubing side of the bead and leave a minimum of one clamp width of hose beyond the clamp. Both the right way and several of the more common wrong ways are shown here. If in doubt or if the clamp is going to be particularly inaccessible, double clamping is a good, low-cost and lightweight insurance. Install the clamps so that you can get a screwdriver or a



Correct relationship of hose, hose clamp and beaded tube.

wrench onto each of them without dismantling anything. Tighten each clamp, pressure test the system and then retighten with the system hot.

Lately I have become fond of the extended wing nut hose clamp developed by the drag racers. This is just one example of the lessons that the drag racers can teach us about doing things in a hurry but doing them right. That is not, however, the reason that I admire these clamps. I like them because even Super Mechanic cannot overtighten them. Most hose clamp failures happen during installation and are due to overtightening—"Tighten it until it feels funny and then back it off a little." The aircraft manuals specify a maximum tightening torque of 15 in-lb for hose clamps. They also state that the clamps should be maintained at this level. This merely means that you should retighten them with the hoses hot and that they should be checked every so often because the synthetic rubber hose material has a tendency to cold flow from under the clamp area. Loss of torque will always be due to this cold-flowing, not from actual backing off of the clamp.

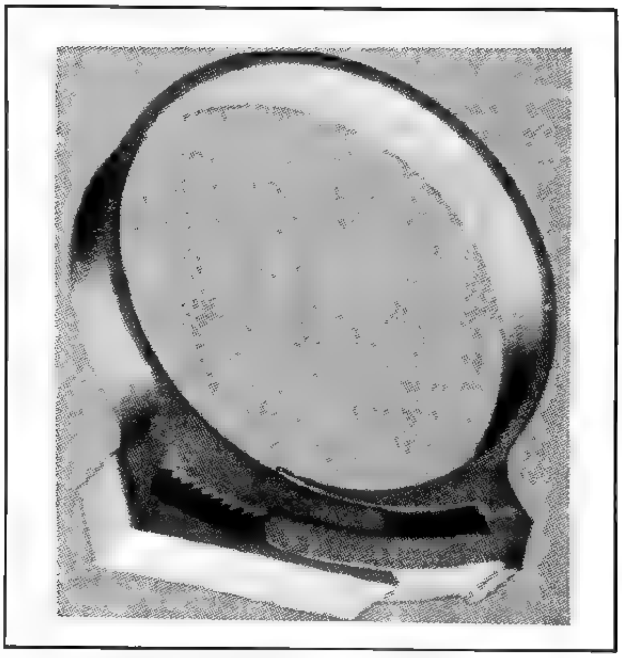
This loss of clamping force due to cold-flow is one of the major reasons why the domestic auto industry changed from the band-type hose clamp to the wire form Corbin clamp. The other reason was cost—both of the clamp and of the labor to install it. I hate the things, but I must admit that they will apply a constant clamping force, regardless of any tendency of the hose to cold-flow. I must also admit that I have never seen failure that I could attribute to a Corbin clamp. I don't use them, though, because they are not adjustable, because they snag parts of my body and clothing on their projecting ends, and because I don't like the way they look.

Another fairly recent development is the toothed, nylon hose clamp. Although they are attractive, compact and light, they scare me simply because I cannot see anyone reaching in to tighten them hot, and there is no convenient way to check them for tightness in service. I don't use them as hose clamps, but they make great locaters for tubular parts like the front wing tube.

Make absolutely sure that no part of the water system, either rubber or metal, can rub against anything at all. This includes the ground, body panels and most especially the moving belts. It is not uncommon to see water lines in close proximity to the exhaust system or to brake lines. In the first case the exhaust heats the water, and in the second the water heats the brake fluid. Another fairly common sin in this area is to arrange things so that the tightening head of a hose clamp is either touching something hard—like a water pump housing—in which case the clamp will fail, or touching something soft—like an adjacent hose—in which case the hose will fail.

I do not believe in the use of stainless steel braid-protected hose for water lines. It is not necessary and it is both expensive and heavy. These, however, are not the major reasons that I do not use it. The real problem is that, in diameters suitable for water hose, metal braid-protected hose is flexible only when compared to hard lines. It is far too stiff to attach to a radiator tube that is soft soldered to a header tank. There is a real possibility of relative motion between the engine and the radiator eventually ripping a water inlet or outlet tube out of the radiator header tank. If someone insists on the appearance of stainless braid, the

Corbin hose clamp.



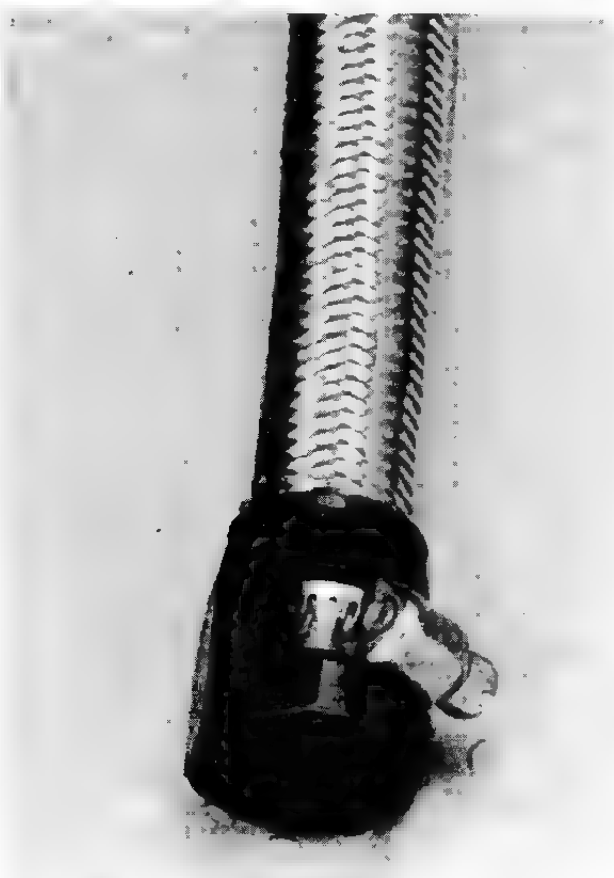
Nylon serrated clamp.

braid itself can be purchased as a trim item and slipped over a standard rubber hose.

If an aluminum anodized hose clamp cover is used, the installation will look like the right stuff and still be perfectly reasonable in both price and performance. Earl's Econ-O-Fit clamp has the advantage of offering some support to the hose itself. The clamp covers are available in just about any color, and the braid is also available in colored nylon. It is well to remember that a metal or nylon cover over an old water hose does not affect the hose itself—it is still just an old water hose.

Oil and fuel lines and fittings

I am frequently asked, "Is all of that lovely and expensive aircraft-style red, blue and silver aluminum and stainless plumbing stuff really necessary?" No, strictly speaking, it isn't really necessary, it's just the best and most economical way to do the job. The advantages of using aircraft specification plumbing hardware are several: It doesn't leak. It doesn't come undone. It doesn't burst, collapse, wear through or wear out. (The stainless steel protective braid will, however, abrade through just about anything that it comes in contact with.) It is highly heat and flame resistant. It is the lightest stuff that will do the job well and conveniently. It is versatile and indefinitely reuseable. If it is not idiotproof, then it is at least fool resistant. It looks nice and it is easy to keep that way.



Trim cover braid and hose clamp cover.

In the remainder of this chapter you are going to read a lot about the Aeroquip Corporation and about Earl's Performance Products. There are excellent reasons for this. The Aeroquip Corporation founded the flexible hose and reuseable hose end industry, and remains the leader in both the aircraft and industrial fields. Earl's Performance Products introduced the hose and hose ends to the world of motor racing and remains the leader there.

Before words like conflict of interest pop into your head, this is probably the time to state that I do not work with the Aeroquip Corporation. I endorse their products simply because of their quality. I *do* work very closely with Earl's, however. I have used their products ever since I returned to this country from Europe in 1965—and I have never had a failure. I do not endorse Earl's products simply because I work with the company; rather, I work with the company because the products are so good that I can endorse them—and because of their many years of direct involvement with racing. Having disposed of that, it is time for a brief history lesson and a short explanation of the innermost workings of the replaceable hose end.

A bit of history

Like most of our really good hardware, our performance plumbing bits originated in the aircraft industry. In the beginning, there was the hose clamp and a natural rubber hose that was wrapped with bicycle tape (in the very beginning, actually, there was a twisted piece of fence wire, once known as the Duesenberg hose clamp). Fluids were mainly conveyed by metal tubing hard lines. The hose and hose clamp gave flexibility where it was needed and provided a convenient method to attach the hard line to both the fluid source and its destination. So long as all that was involved in fluid transmission was gravity fed to a low-pressure carburetor, water hoses to and from a radiator, and (maybe) some low-pressure oil lines, the right hose clamp was satisfactory—after those involved had learned to bead the ends of the hard lines. In fact, the hose clamp was in all probability better than the hose that was available. In most passenger car applications the right hose clamp is still perfectly satisfactory.

As military and commercial aircraft became more sophisticated, lubricants had to be conveyed over longer distances and at higher pressures, thus the high-pressure fuel-injection system came into being. Retractable landing gear, variable-pitch propellers, flaps, bomb bay doors, gun turrets and the like were developed and actuated by hydraulic sys-

tems. In short, the conveyance of various fluids became both more complex and more critical.

At the same time the increased weight and increased landing speeds of the aircraft (as well as the effect on the paying passenger) made it impractical to continue the practice of putting down in a convenient cow pasture to effect a quick repair. The pressures, temperatures and number of cycles involved rendered the old faithful hose clamp obsolete. Neither were the old hoses up to the new tasks. Extensive use was, and is, made of hard lines or rigid metal tubing. But, due to the articulation of the actuated parts and to the danger of vibration, there was a need for flexible hydraulic hoses. One of the needs that had to be met was the crucial requirement that mechanics in the field (and I mean in the back of beyond, not the airfield as in LAX) be able to repair or replace damaged components from a minimal stock of parts and with common hand tools—while being shot at. In those days we still knew how to make equipment with which to fight a war.

A whole generation of lightweight and flexible medium-pressure hose and replaceable hose ends was developed to meet the need. The hoses were connected to fluid reservoirs, pumps, motors and the like by means of reuseable hose ends and a wide range of threaded adaptors. The hose ends threaded onto the adaptors and sealed through matching machined cones. The area of the cones and the residual stress in the threads was sufficient to positively lock the assembly. The hose ends were manufactured in straight, 45 degree and 90 degree styles, and the adaptors were manufactured in seemingly endless configurations, some of which are pictured here.

Through innovative design and engineering excellence, the Aeroquip Corporation became the world leader in the design and manufacture of flexible hoses and replaceable hose ends. Sizes and configurations were standardized to match the AN system of hard line sizes. From the original three configurations, hose ends have multiplied like rabbits. Because of their specialized interest in racing, Earl's innovative technology has resulted in a selection of hose end configurations.

Plumbing size designations

A fair bit of confusion has been expressed regarding the AN, MS and NPT size designations used in both race car and aircraft plumbing. This is not surprising; the system seems to have been designed to cause confusion. We will attack the AN system first.

AN tubing sizes

The AN (Air Corps/Navy) standards were established prior to World War II in order to organize and standardize the sizes, configurations and specifications of military aircraft hardware into a servicewide system of interchangeable standard parts. The system soon became industrywide. It has been greatly expanded in the ensuing decades and has gone through a couple of name changes. With respect to plumbing, the number that we usually refer to as the AN number is actually a size designation and comprises only a part of the full AN identifying number. It is properly referred to as the AN dash number, and does not define the inside diameter of a flexible hose. Instead it refers to the outside diameter, in increments of 1/16 in., of the metal hard line that is considered to be the equivalent (in flow rate) of the flexible hose. The hard line outside diameter or O.D. was used as the standard simply so that the metal flared tube compression sleeves and coupling nuts used with hard lines could be standardized.

Accordingly, each AN tube diameter was assigned a corresponding thread diameter and pitch for its coupling members. This thread designation refers to the threads of the hard line coupling nut, the flexible hose end coupling nut and the AN adaptors that go with them. Thus any AN dash 10 coupler will fit and seal on any AN dash 10 male adaptor. This does not, however, mean that *any* dash 10 hose end can be used with *any* dash 10 hose. Hopefully, the table and the thread silhouettes will clarify the situation.

Pipe threads

For many years now I have considered the tapered pipe thread to be an abomination. But this opinion, no matter how valid it may be, has done nothing toward the abolishment of the tapered pipe thread. We are, it seems, stuck with the damned things—so we may as well learn to identify them. National Tapered Pipe (NTP) numbers refer to the inside diameter of the piece of heavy wall plumbers pipe that was originally designed to receive the male thread. It is sort of like our inch system of measurement; the king whose foot was twelve times the distance between the first and second joints of his index finger is long dead, but we are stuck with the system.

Anyway, the threads themselves are so designed that the male thread is an interference fit in its female counterpart, and it is that interference that keeps the drain lines and some of the water supply lines in your house from leaking—usually, that is. Unfortunately a great number of the castings that we use come equipped with female pipe threads. Some of the AN adaptors are manufactured with tapered pipe threads on one end and AN threads and sealing cones on the other. Although I flatly refuse to use pipe threads on high-pressure systems, I must admit that I am usually too lazy to change them on existing fuel and oil systems—so long as I do not intend to remove the adaptor from its female housing very often. The use

of Teflon (either in tape or paste form) is an absolute necessity with pipe threads. The earlier illustration also shows how the size designation system works.

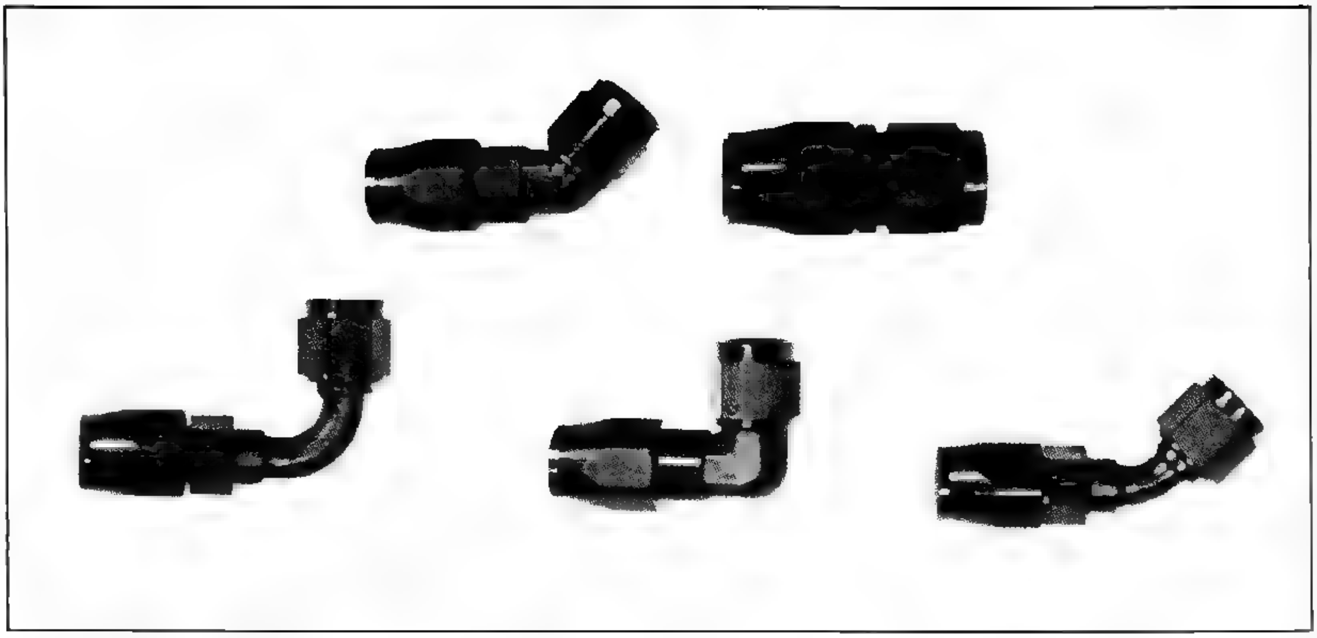
British castings come, naturally enough, with British Straight Pipe (BSP) threads. BSP thread pitches are close enough to NTP so that running the appropriate NTP tap through the BSP hole works pretty well. Alternatively, both Earl's Performance Products and Torino Motor Racing make a range of BSP to AN adaptor fittings. Aeroquip manufactures a line of BSP hose ends, but only for industrial hose (the same is true of metric).

Single-nipple replaceable hose end

The early hoses were flexible only in comparison to the hard lines that they replaced. They featured thick walls of synthetic rubber reinforced with both external and internal braided protective sheaths of fabric and/or metal, which both protected the hose from abrasion and increased its pressure capacity. The hose ends were of the single nipple configuration.

In use, the female threaded socket is slipped over the cut end of the hose and the male threaded nipple is threaded into the socket—and into the hose liner. As the tapered nipple advances into the threads of the socket, the hose is wedged firmly into the annular space between the I.D. of the socket and the O.D. of the nipple. The interior shape of the socket is designed to receive and wedge the hose and is provided with annular barbs both to prevent the hose from backing out of the socket during assembly and to help to retain the hose on the hose end under pressure. This wedging of the hose between the socket and the nipple both retains the hose end and provides the primary—in this case the only—seal for the assembly.

Early, heavy wall metal braid reinforced flexible hose and single-nipple reuseable hose end.



Straight, 45 degree and 90-degree hose ends.

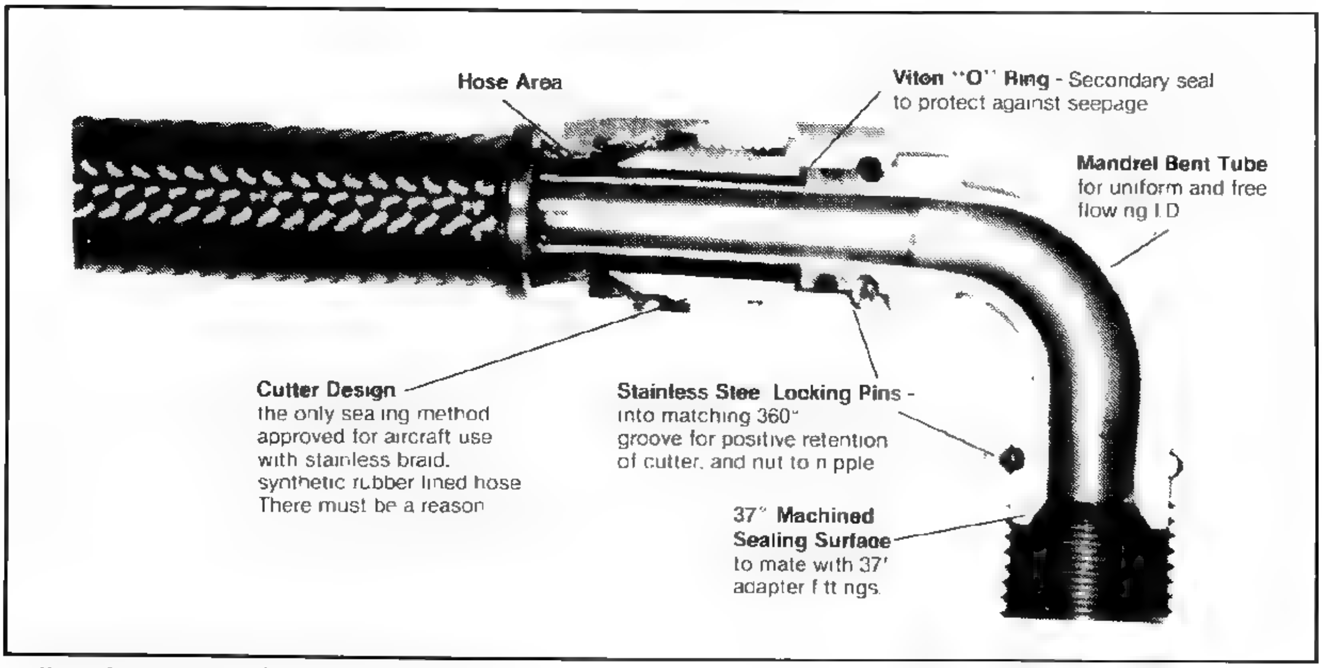
The original single-nipple hose end is a relatively inexpensive part to manufacture and it is perfectly suitable for use with the relatively thick-walled hose that it was designed for. In many industrial applications, where weight and extreme flexibility are not factors, it is still the most popular style—especially if the hose I.D. is greater than ⅝ in. In the smaller sizes the necessarily heavy wall of the nipple reduces the cross-sectional area of the assembly sufficiently to cause significant reductions in flow capacity. This reduction is not particularly important in many of our applications. However, the fact that the single-nipple hose end, when assembled on modern, thin-walled flexible hose, is liable to leak (or even to blow off the hose) under pressure *is* of considerable significance and concern. Read on!

The complexity of the modern aircraft increased at a greater rate than did the power available to make them fly, and the importance of component weight changed from critical to super-critical. Advances in both synthetic rubber technology and in manufacturing techniques led to the development and almost universal use of ultralight and extremely flexible low- and medium-pressure hose of very thin walled synthetic rubber. This hose is protected against both abrasion and hernia by a partial stainless steel braided sheath molded into the hose liner, and a full-coverage stainless wire braided outer cover bonded to the hose.

The Aeroquip Corporation led the way. During the development phase of this type of hose, it was discovered that the traditional single-nipple hose end could not be successfully adapted for reliable use. The basic problem was that, if the hose end was designed so that it would compress the hose sufficiently to positively prevent blowoff, the soft inner liner was to be liable to cold flow under compression. Sometime after assembly, this cold flowing would reduce the thickness of the portion of the hose that was wedged between the nipple and the socket. If fortune smiled, the assembly would begin to leak. If fortune did not smile, the hose end would blow off the hose. Conversely, if the initial compression was reduced to the point where the hose liner would not cold flow, neither a positive seal nor a secure hose retention could be ensured.

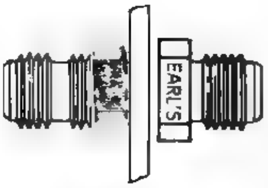
Double-nipple hose end

The solution was the nipple and cutter or double-nipple hose end—which I reckon to be one of the more clever devices of the last half century. In this design, a threaded socket is slipped over the cut end of the hose and the male threaded nipple is screwed into it just as in the single-nipple hose end. The difference lies in the detail design of the nipple itself—a deceptively complex part. This nipple is, in effect (and sometimes in fact), two separate parts, hence the name. Just as with the single nipple, the nipple tube slips inside the hose liner, but it is not threaded. Concentric with the nipple tube, and attached to it, is the sharp-edged cutter. An annular chamber of closely controlled dimensions is formed between the inside surface of the cutter and the outside surface of the nipple tube. The outside of



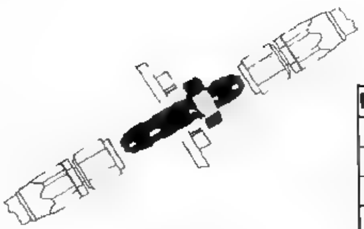
Earl's Performance Products Swivel Seal hose end.

A.N. to A.N. Adapters, Continued



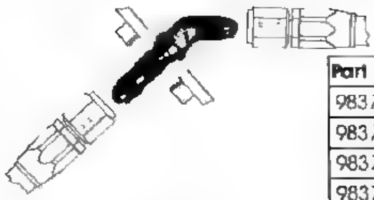
BULKHEAD NUT

Part No.	Qty. Per Pkg.	Ftg. Size
592403	2	3
592404	2	4
592406	2	6
992408	1	8
992410	1	10
992412	1	12
992416	1	16



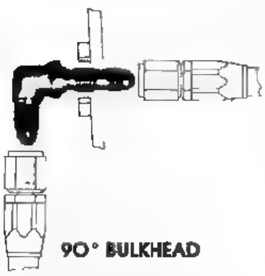
STRAIGHT BULKHEAD

Part No.	Ftg. Size
983203	3
983204	4
983206	6
983208	8
983210	10
983212	12
983216	16



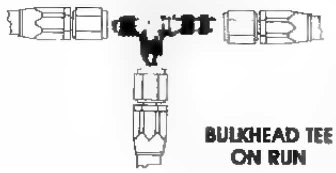
45° BULKHEAD

Part No.	Ftg. Size
983703	3
983704	4
983706	6
983708	8
983710	10
983712	12



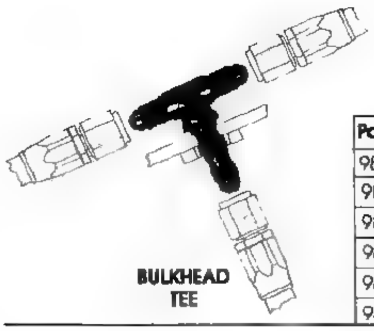
90° BULKHEAD

Part No.	Ftg. Size
983303	3
983304	4
983306	6
983308	8
983310	10
983312	12



BULKHEAD TEE ON RUN

Part No.	Ftg. Size
980404	4
980406	6
980408	8

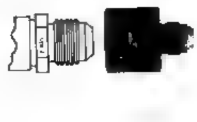


BULKHEAD TEE

Part No.	Ftg. Size
983403	3
983404	4
983406	6
983408	8
983410	10
983412	12

FEMALE A.N. TEE

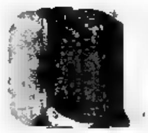
Part No.	Ftg. Size
993804	4
993806	6
993808	8
993810	10
993812	12



AN REDUCER (O-RING SEAL)

Part No.	Female Ftg. Size	Male Ftg. Size
989404	10	8
989409	12	8
989410	12	10
989411	16	12
989443	4	3
989464	6	4
989486	8	6

TUBE NUT

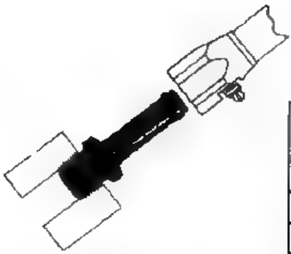


Part No.	Qty. Per. Pkg.	Ftg. Size	Tube Size
581803	2	3	3/16
581804	2	4	1/4
581806	2	6	3/8
981808	1	8	1/2
981810	1	10	5/8
981812	1	12	3/4
981816	1	16	1

TUBE SLEEVE

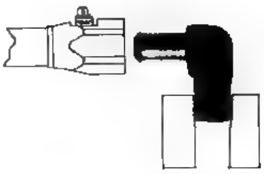
Part No.	Qty. Per Pkg.	Ftg. Size	Tube Size
581903	2	3	3/16
581904	2	4	1/4
581906	2	6	3/8
981908	1	8	1/2
981910	1	10	5/8
981912	1	12	3/4
981916	1	16	1

Hose Barb to Pipe Thread Adapters



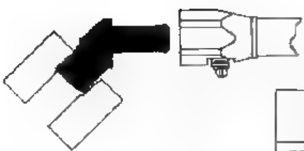
STRAIGHT

Part No.	Hose I.D. Size	Pipe Thd. Size
984004	1/4	1/8
984006	3/8	1/4
984008	1/2	3/8
984010	5/8	1/2
984012	3/4	3/4



90°

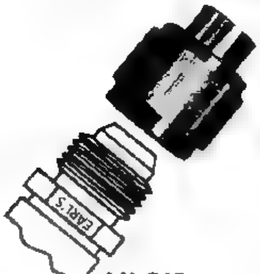
Part No.	Hose I.D.	Pipe Thd. Size
984204	1/4	1/8
984206	3/8	1/4
984208	1/2	3/8
984210	5/8	1/2
984212	3/4	3/4



45°

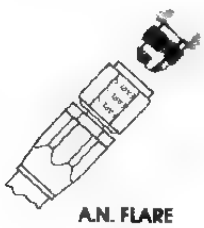
Part No.	Hose I.D. Size	Pipe Thd. Size
984404	1/4	1/8
984406	3/8	1/4
984408	1/2	3/8
984410	5/8	1/2
984412	3/4	3/4

Caps & Plugs



A.N. CAP

Part No.	Qty. Per Pkg.	Ftg. Size
592903	2	3
592904	2	4
592906	2	6
992908	1	8
992910	1	10
992912	1	12
992916	1	16

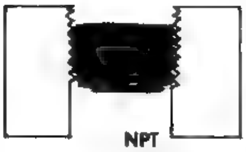


A.N. FLARE PLUG

Part No.	Qty. Per Pkg.	Ftg. Size
580603	2	3
580604	2	4
580606	2	6
980608	1	8
980610	1	10
980612	1	12
980616	1	16

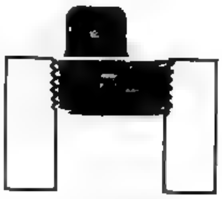
PLUG PORT

Part No.	Qty. Per Pkg.	Ftg. Size
581403	2	3
581404	2	4
581406	2	6
981408	1	8
981410	1	10
981412	1	12
981416	1	16



NPT ALLEN HEAD PLUG

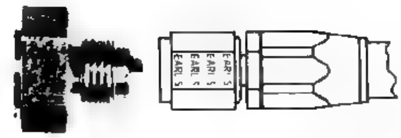
Part No	Qty. Per Pkg.	Pipe Thd. Size
593201	2	1/16
593202	2	1/8
593203	2	1/4
593204	2	3/8
993205	1	1/2
993206	1	3/4



NPT PLUG - SQUARE

Part No.	Pipe Thd. Size
991301	1/8
991302	1/4
991303	3/8
991304	1/2
991306	3/4

Aluminum Weld Fittings



MALE "AN"

Part No.	Ftg. Size
997106	6
997108	8
997110	10
997112	12
997116	16

FEMALE "AN"

Part No.	Ftg. Size
987104	4
987106	6
987108	8
987110	10
987112	12
987116	16

FEMALE NPT

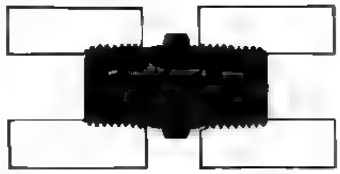
Part No.	Thd Size
996701	1/8" NPT
996702	1/4" NPT
996703	3/8" NPT
996704	1/2" NPT
996706	3/4 NPT

Pipe Thread Adapters

COUPLING

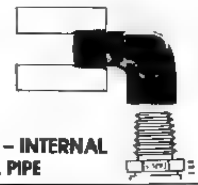
Part No.	Pipe Thd. Size
991001	1/8
991002	1/4
991003	3/8
991004	1/2
991006	3/4

Pipe Thread Adapters, Continued



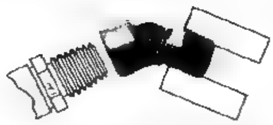
NIPPLE

Part No.	Pipe Thd. Size
991101	1/8
991102	1/4
991103	3/8
991104	1/2
991106	3/4



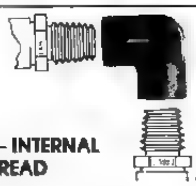
90° ELBOW – INTERNAL EXT. PIPE

Part No.	Pipe Thd. Size
991401	1/8
991402	1/4
991403	3/8



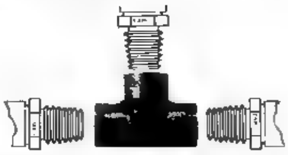
45° ELBOW – INTERNAL EXT. PIPE

Part No.	Pipe Thd Size
991501	1/8
991502	1/4
991503	3/8



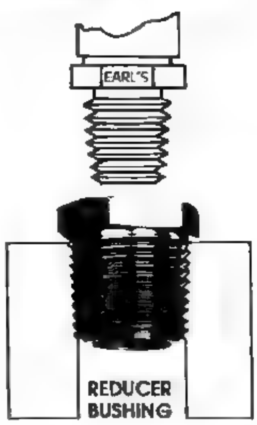
90° ELBOW – INTERNAL PIPE THREAD

Part No.	Pipe Thd. Size
991601	1/8
991602	1/4
991603	3/8



TEE-INTERNAL PIPE THREAD

Part No.	Pipe Thd. Size
991701	1/8
991702	1/4
991703	3/8



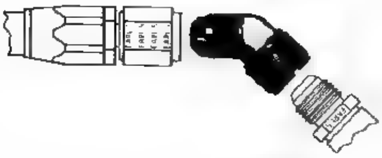
REDUCER BUSHING

Part No.	Female Pipe Thd Size	Male Pipe Thd. Size
991201	1/8	1/4
991202	1/4	3/8
991203	1/8	3/8
991204	3/8	1/2
991205	1/4	1/2
991206	1/8	1/2
991207	1/2	3/4
991208	3/8	3/4
991209	1/4	3/4
991210	3/4	1
991211	1/2	1
991212	3/8	1
991213	3/4	1¼

Special Purpose Adapters

STRAIGHT FEMALE AN SWIVEL COUPLING

Part No.	Hose End Size
915104	4
915106	6
915186	8 6
915108	8
915110	10
915112	12
915116	16

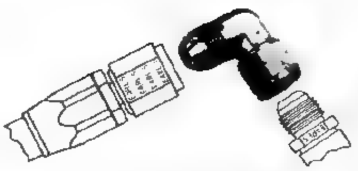


45 FEMALE AN SWIVEL TO MALE AN

Part No.	Hose End Size
924104	4
924106	6
924108	8
924110	10
924112	12

FEMALE AN SWIVEL TO MALE PIPE THD.

Part No.	Hose End Size	NPT Thread Size
916106	6	1/4
916162	6	1/8
916166	6	3/8
916107	8	1/4
916108	8	3/8
916188	8	1/2
916110	10.	1/2
916111	10	3/8
916112	12	3/4
916113	12	1/2



90 FEMALE AN SWIVEL TO MALE AN

Part No.	Hose End Size
921104	4
921106	6
921108	8
921110	10
921112	12

90 MALE AN TO MALE SWIVEL PIPE THD.

Part No.	Hose End Size	NPT Thread Size
922106	6	1/4
922162	6	1/8
922166	6	3/8
922107	8	1/4
922108	8	3/8
922188	8	1/2
922110	10	1/2
922111	10	3/8
922113	12	1/2
922112	12	3/4

90 FEMALE AN SWIVEL TO MALE SWIVEL PIPE THD.

Part No.	Hose End Size	NPT Thread Size
923106	6	1/4
923162	6	1/8
923166	6	3/8
923107	8	1/4
923108	8	3/8
923188	8	1/2
923110	10	1/2
923111	10	3/8
923113	12	1/2
923112	12	3/4

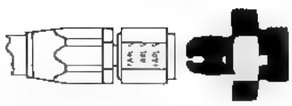
T-FITTING FEMALE SWIVEL ON SIDE

Part No.	Hose End Size
925104	4
925106	6
925108	8
925110	10

T-FITTING FEMALE SWIVEL ON RUN

Part No.	Hose End Size
926104	4
926106	6
926108	8
926110	10

Carburetor & Fuel Pump Adapters



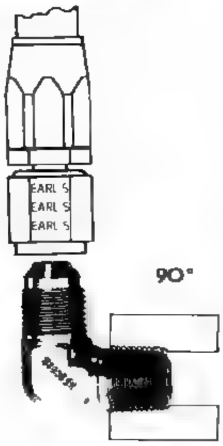
CARB ADAPTERS

Part No.	Carb Thd. Size	Hose End Size	Recommended Application
991941	5/8 20	6	Carter
991942	9/16 24	6	Single Inlet Holley
991943	7/8 20	6	Dua Feed Holley
991944	12mm x 1 50	6	Weber
991945	12mm x 1 25	6	Deiiorto/Solex
991946	1/2-20	6	Power Steering/Fuel Pump
991947	5/8 18	6	Power Steering/Fuel Pump
991948	7/8-20	8	Dual Feed Hol ey
991949	7/8 20	6	Rochester
991950	11/16-16	6	Power Steering
991951	7mm Banjo Bolt	6	Dellorto
991952	1" 20	6	Rochester 75 up

Pipe Thread to A.N. Adapter

STRAIGHT

Part No.	Ftg. Size	Pipe Thd. Size
981603	3	1/8
981604	4	1/8
981644	4	1/4
981606	6	1/4
981662	6	1/8
981666	6	3/8
981688	8	1/2
981607	8	1/4
981608	8	3/8
981610	10	1/2
981611	10	3/8
981612	12	3/4
981613	12	1/2
981615	16	3/4
981616	16	1

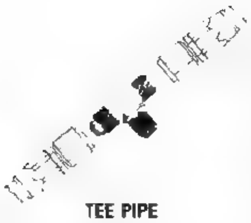


90°

Part No.	Ftg. Size	Pipe Thd. Size
982203	3	1/8
982204	4	1/8
982244	4	1/4
982206	6	1/4
982262	6	1/8
982266	6	3/8
982208	8	3/8
982288	8	1/2
982210	10	1/2
982212	12	3/4
982213	12	1/2
982215	16	3/4
982216	16	1

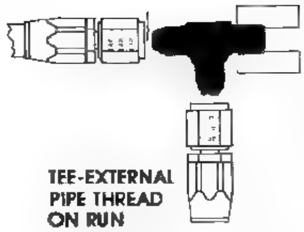
45° ELBOW

Part No.	Ftg Size	Pipe Thd Size
982303	3	1/8
982304	4	1/8
982306	6	1/4
982308	8	3/8
982310	10	1/2
982312	12	3/4
982316	16	1



TEE PIPE THREAD ON SIDE

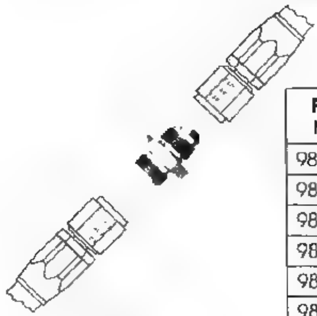
Part No.	Ftg. Size	Pipe Thd. Size
982503	3	1/8
982504	4	1/8
982506	6	1/4
982508	8	3/8
982510	10	1/2
982512	12	3/4



TEE-EXTERNAL PIPE THREAD ON RUN

Part No.	Ftg. Size	Pipe Thd. Size
982603	3	1/8
982604	4	1/8
982606	6	1/4
982608	8	3/8
982610	10	1/2
982612	12	3/4

A.N. To A.N. Adapters

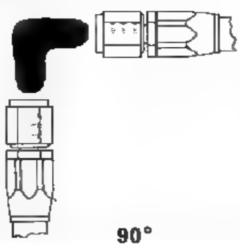


UNION

Part No.	Ftg. Size
981503	3
981504	4
981506	6
981508	8
981510	10
981512	12
981516	16

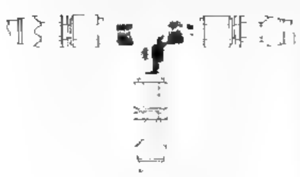
REDUCER

Part No.	Large Ftg. Size	Small Ftg. Size
991902	4	3
991906	6	4
991912	8	6
991914	10	6
991915	10	8
991918	12	6
991919	12	8
991920	12	10
991922	16	10
991923	16	12



90°

Part No.	Ftg Size
982103	3
982104	4
982106	6
982108	8
982110	10
982112	12
982116	16



TEE

Part No.	Ftg Size
982403	3
982404	4
982406	6
982408	8
982410	10
982412	12
982416	16

AN adaptor fitting chart.

the cutter has male threads that match the female threads of the socket.

As the nipple/cutter assembly advances into the socket threads, the sharp leading edge of the cutter separates the soft inner hose liner into an inner tube and an outer tube. The resulting inner tube, which is all synthetic rubber, is captured (but not compressed) inside the annular chamber between the nipple tube and the cutter to form the primary seal of the assembly. The outer tube and both layers of protective braid are wedged into the space between the outside of the nipple/cutter assembly and the inside of the socket—both of which are designed and shaped to ensure positive hose retention.

Thus the sealing function and the hose retention function are accomplished separately. Since the contained fluid is not in contact with the threads of either part, there is no spiral leak path and there can be no seepage past the threads. As the portion of the nipple that slips inside the hose is not threaded, it can be designed with a very thin wall, and flow restriction through the assembly is minimized. Very clever—it cannot have been easy to develop.

In the last years of World War II and again during the Korean War, the high-tech plumbing was turned out by the shipload and soon found its way onto the surplus market. About that time, Earl Fouts decided to set up Earl's Performance Products as an aircraft surplus house in Gardena, California. At that time Gardena was one of the centers of Indy car racing. In fact, one of Earl's neighbors was Quinn Epperly, and George Bignotti was in and out of Epperly's frequently. One thing led to another and soon Bignotti's Indy cars sprouted surplus aircraft hose. The rest of the Indy contingent wasn't far behind. Lance Reventlow's Scarabs followed suit and soon everybody who was anybody in racing was plumbing their cars with surplus hose, hose ends and adaptors from Earl's. The trend soon spread to boats and hot rods. In some ways it turned out that Earl had grabbed a tiger by the tail.

Several years later, Robert MacNamara became Secretary of Defense and tried to run the DOD as he had the Ford Motor Company (for the better, in my opinion—at least he realized that they were wasting billions of dollars and tried to do something about it). Cost plus military contracts became a thing of the past (only to be replaced by cost override clauses) and the surplus market dried up. Earl's contacted the Aeroquip Corporation and asked for a distributorship. Aeroquip did not consider the automotive high-performance/racing market to be worthwhile (talk about misreading a market!) and refused to sell to lowly surplus dealers. Earl's Performance Products was faced with the choice of either abandoning the market that they had established and developed, or of designing and manufacturing their own line of hose and hose ends. Fortunately for the racers, they decided to bite the bullet and become manufacturers.

The hose was not a problem, as several companies were producing metal-braid-protected hose to military specs and were as willing to sell to Earl's as they were to Aeroquip or anyone else. Neither Earl nor his son Bob thought that it made sense to go to the effort and expense necessary to design and produce a hose end if the end result would be a product only as good as what was already available. So they set out to improve upon the recognized standard of a very sophisticated industry.

Against all odds they succeeded. Earl's Swivel Seal hose end is, for racing purposes, superior to Aeroquip's Little Gem. The superiority is not due to increased reliability, for they are each as reliable as a mechanical device can be. Nor is the superiority due to decreased cost; the difference in price is minimal. One advantage to buying the Earl's hose end is that they offer more configurations and better service than Aeroquip. But the major advantage lies in the fact that Earl's got very clever indeed and improved upon the original design of the double-nipple hose end. They made the Swivel Seal radially adjustable with respect to the hose after final assembly. This does away with the previously necessary (and frustrating) process of clocking angled hose ends so that they would end up pointing in the required direction. If this wasn't done—and done correctly—the hose ends could not be installed onto their respective adaptors without putting a torsional preload or even a kink in the hose. Or, in the case of short hose runs, they could not be installed at all.

There are two possible dangers attendant to a hose with improperly clocked hose ends. First, a torsionally preloaded or twisted hose is considerably weakened and prone to aneurysm. Second, under just the wrong combination of conditions, a torsional preload can force the hose to attempt to unwind like a twisted rubber band so that the hose end becomes self-loosening on the adaptor.

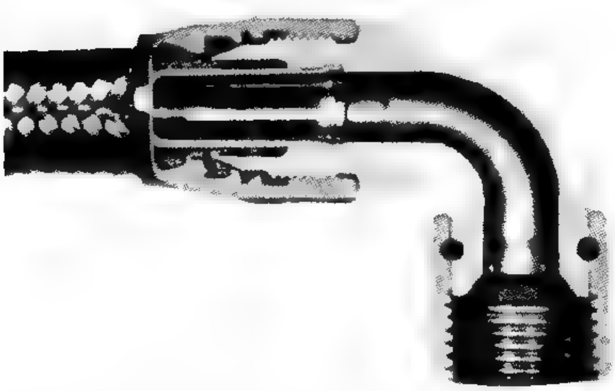
By the simple but brilliant expedient of designing the cutter so that it is positively sealed to and retained to the nipple, but is free to rotate on it, Earl's has done away with all that. This may not sound like a big deal, but take my word for it, to those of us who make up a lot of hoses, it is a very big deal. This description of the inner workings of the Swivel Seal, which I wrote for their catalog some years ago, is reprinted with Earl's permission.

"The SWIVEL SEAL is supplied pre-assembled from the four major components illustrated—a socket, a cutter, a nipple and a coupling member. In this illustration the coupler is a female 'B' nut which threads onto and seals against standard Earl's

Supply, AN and MS 37 degree male cone adaptor fittings. The 'B' nut is free to rotate on the nipple but is retained to it by a pin which is inserted through a hole in the nut and bent into a matching 360 degree groove machined into the nipple. When the 'B' nut is tightened onto the adaptor fitting the female seat of the nipple seals against the male cone of the adaptor. The threads have no sealing function.

"A Viton 'O' ring is placed in the appropriate groove on the hose end side of the nipple. The cutter, which is machined with a matching 'O' ring recess, is then pushed over the nipple and retained on it by a pin in the same way that the coupler is retained on the other end of the nipple. The cutter is now sealed to and retained on the nipple—*but can be rotated with respect to it.* This feature is unique (and patented). The assembly of the cutter on the nipple leaves an annular chamber between the cutter and the outer surface of the nipple tube. The nipple tube protrudes well beyond the sharp edge of the cutter.

"To assemble the hose end onto a hose, the socket is first slipped over the end of the hose and the nipple tube is inserted into the nitrile inner hose liner until the cutter edge butts against the liner. Threading the cutter into the socket then forces the inner tube onto the edge of the cutter which separates the liner into an inner tube of synthetic rubber and an outer tube which includes the intermediate partial coverage reinforcing braid. The inner tube is forced into the annular chamber between the I.D. of the cutter and the O.D. of the nipple tube and there forms the primary seal of the assembly. The outer tube, along with both the inner and the outer protective braid, is captured between the O.D. of the cutter and the I.D. of the socket—both of which are shaped for the purpose. This controlled wedging action retains the hose end onto the hose. The 'O' ring between the cutter and the nipple provides a secondary or 'anti-seepage' seal. The design thus provides positive hose retention and a positive seal—with the sealing and retention functions separated. This 'double nipple' configuration hose end offers both more positive seal and stronger retention than either the conventional 'single nipple' or the 'push on' type. In addition, since the cutter can be rotated on the nipple, the nipple and cutter can be rotated with respect to the hose after assembly. This feature is unique to the SWIVEL SEAL™ hose end and allows angled hose ends to be aligned or 'clocked' to suit the requirements of the individual assembly without either disassembly or overtightening . . ."



Aeroquip Corporation single-nipple hose end for use on lightweight hose.

When the racing/high-performance market proved to be considerably larger than they had predicted, Aeroquip, caught in a recession, decided that our money was green after all and, mainly through their industrial distributors, started a campaign aimed at the high-performance auto/boat aftermarket industry.

Predictably, the market has attracted some new manufacturers—and some pretty heavy hype. Mainly, the newcomers are mass merchandising inferior products that look like the right stuff—but are not. To my knowledge only Earl's and Aeroquip currently manufacture double-nipple hose ends for use with lightweight, thin-wall stainless steel braid-protected hose. One of the hypsters has been spending a lot of money on an advertisement trying to convince the unwary and the uninformed that both Earl's and Aeroquip (to say nothing of the US Navy Bureau of Aeronautics, the US Air Force and the FAA) are *wrong.* The hypster claims that not only is the single-nipple hose end emminently suitable for use with lightweight hose, but that the assembly of the double-nipple hose end often leads to the inadvertent creation of a flapper or accidental one-way valve and that therefore the single nipple is the only way to go. God knows that it is possible to make a flapper in a double-nipple hose end assembly—but you really have to work at it. I should also point out that this particular ad is technically incorrect: when a flapper is created, it is the inner nipple that does the dirty, not the cutter.

In all of the years that I have been using double-nipple hose ends, I have seen exactly one flapper—and that one was made by a very competent man who just got careless once. Of course, being competent, he did find it before it got onto a car (by the simple expedient of trying to pass clean solvent through the assembly in both directions as a final test before installation). The double-nipple configuration of hose end is the only type currently approved for use with lightweight hose by the people who refuse to take *any* chances with equipment—the US Navy, the US Air Force and the FAA. There has to be a message there someplace.

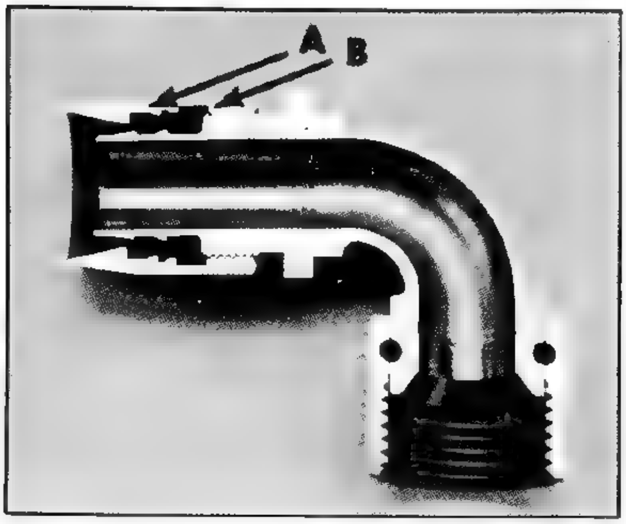
Recent developments in the single-nipple scene

I am not saying that it is not *possible* to design an alternative hose end configuration that is suitable for use with lightweight hose. Western Coupling came close a few years ago with a modified push-on barb-style hose end—but it seeped and was reuseable only with difficulty. It wasn't particularly well marketed and I haven't seen one in years.

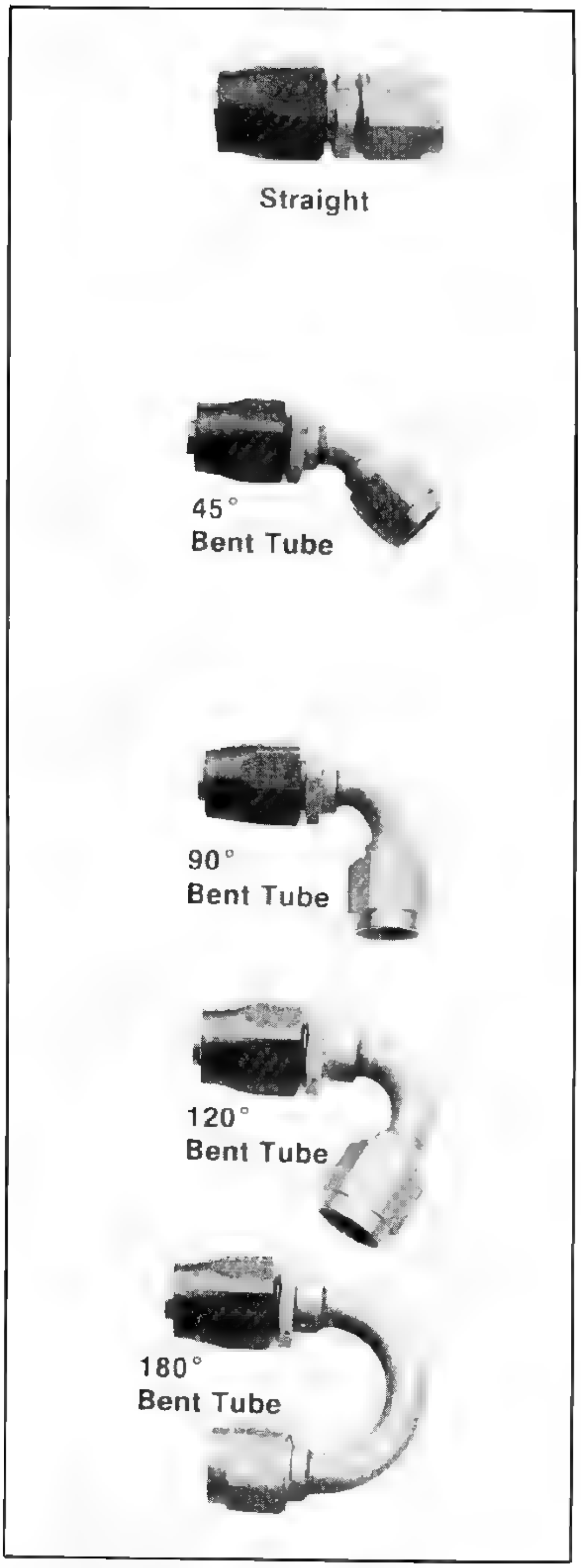
There have been some significant advances in the compounding of the elastomers used in the hoses in the past few years, and they are now considerably less subject to cold flowing than they used to be. Neither work hardening nor heat degradation are a problem with the new elastomers used by both Earl's and Aeroquip—which are rated for continuous use at temperatures.

When the Aeroquip Corporation figured out in the late 1970s that the high-performance aftermarket was worth going after, they came out with a single-nipple hose end designed specifically for use with their lightweight engine hose. This hose end is not approved for aircraft use but is perfectly adequate for most automotive applications. I have never heard of a properly installed Aeroquip single-nipple hose end blowing off the hose, although like all conventional single-nipple designs, they can weep a little bit through the threads which serve as both seal and retention. The other manufacturers' single-nipple hose ends should be avoided like the plague.

After several years of development, Earl's recently introduced their Auto-Fit hose end. This fitting represents a new concept in the design of single-nipple hose ends. It combines the threaded socket and wedge-shaped hose retention area of the conventional single-nipple hose end with the



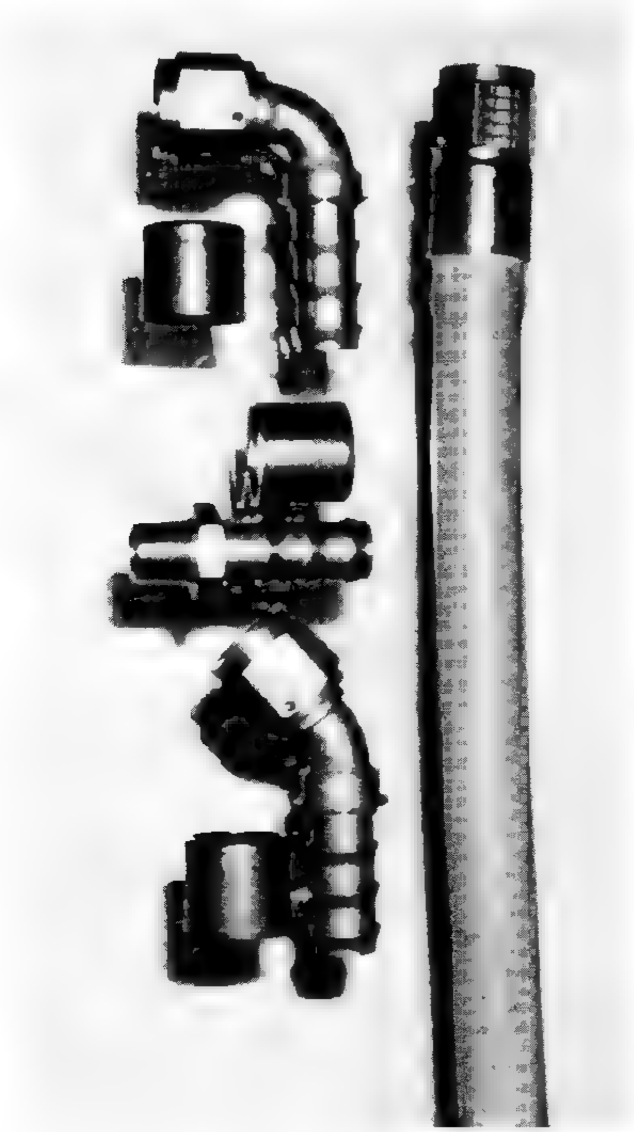
Earl's Auto-Fit hose end creates seals at both point A and B. Other single-nipple designs use threads to create the seal, forming a path for seepage.



Earl's Performance Products Auto-Fit single-nipple hose end for use on lightweight hose.

barbed nipple of the push-on hose end. In this configuration the seal is effected by compression of the hose against the annular barbs (labeled A and B in the illustration). The compression is achieved by the wedging action of the threaded socket, whose interior shape is identical to that of the Swivel Seal. The hose retention function is carried out both by the barbs and by controlled wedging of the hose and braid between the nipple barbs and the socket. Unlike the conventional single-nipple hose end, the threads are not part of the seal and the amount of crush necessary to ensure hose retention is reduced below the point where there is any danger of cold-flow.

Earl's designers are cautious people; this particular hose end was in development for more than two years. It has been through a lot of testing. It works. In its final configuration there have been no failures and no seepage. The Auto-Fit hose end is, in fact, as safe and as secure as the Swivel Seal. It is not radially adjustable after assembly, but they are working on that. Since it is a single-nipple design, its inside diameter is slightly less than that of the same size Swivel Seal so there is some flow reduction—but only in comparison to the nipple and cutter hose ends. It is presently available in straight, 45 degree, 90 degree, 120 degree and 180 degree configurations for hose sizes from dash four through dash twenty. The Auto-Fit is obviously less expensive to manufacture than the Swivel Seal, and Earl's Performance Products is passing the savings along to us: the Auto-Fit is considerably less expensive than the corresponding Swivel Seal.



Earl's Super Stock hose and hose ends.

Push-on or barb and clamp hose end

These high-tech hose ends are all too expensive for the automotive industry—and, in all honesty, they are not necessary. The traditional push-on barb and clamp hose end works just fine for almost all normal (for instance, nonracing and relatively low-pressure) automotive and marine applications. The hose is pushed on over the annular bulb or barbs on the male hose end. A conventional clamp is installed upstream of the last barb and that is all there is to it.

The system may lack sophistication and, in its original form, it is not very pretty, but it has worked well for quite a long time. In the case of the bulb, you disassemble by removing the clamp and pulling off the hose. In the case of the barbs, you disassemble by cutting the hose. The hose ends are reuseable.

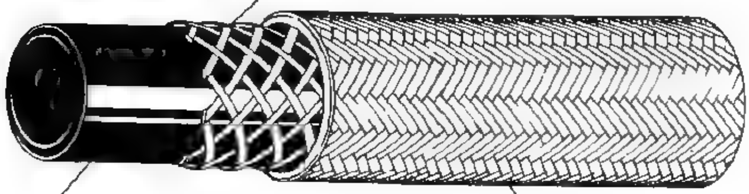
This type of hose end is inexpensive to manufacture, and several companies are now marketing them in plated steel, brass or in anodized aluminum. Earl's has a particularly elegant line which they call their Super Stock series. Combined with an anodized nipple cover, they are an attractive, reliable and inexpensive alternative to the right stuff for most nonflying and nonracing applications. The barbs and bulbs are designed to be used with specific types of hose, and you should respect the manufacturer's recommendations when selecting either hose or hose ends.

Moonglow

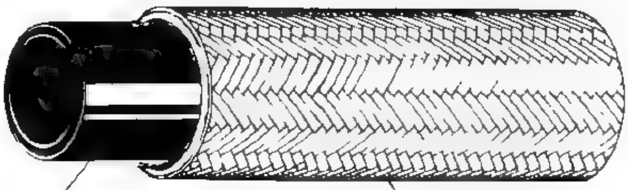
There is an old saying that there is nothing that someone cannot make cheaper so long as they are willing to make it worse. This is particularly true of hose ends. Further, the inferior stuff look just about like the right stuff—at least to the untrained eye.

To illustrate, the picture shows hose ends from three different manufacturers. Very similar, right? In comparison, the second picture shows cut-away views of the same three hose ends. In the first view, Aeroquip's single-nipple style, the hose is firmly wedged between the socket and the nipple. The same is even more true of Earl's Auto-Fit hose end in the third view. However, the design of the socket is such that the hose is nowhere near as firmly

A. AIRCRAFT AND RACING SPEC HOSE (EARL'S PERFORM-O-FLEX™)



B. HIGH-PERFORMANCE STREET HOSE (EARL'S ECON-O-BRAID™)



C. MOONGLOW

Cut-away view of three grades of stainless-braid reinforced flexible hose.

retained. This particular brand of hose end, in fact, is rather well known in racing for blowing off the hose.

A photo of a test specimen of Earl's ES 400 hose and an Auto-Fit single-nipple hose end is also shown. The hose failed at 3,000 psi above its rated pressure. There was no detectable deterioration of the seal between the hose and the hose end. This is typical of the right stuff.

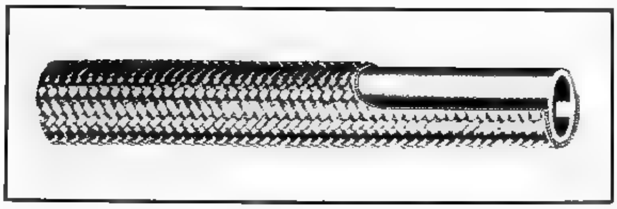
My personal rules on high-performance plumbing of fuel and oil systems are simple. I will use Earl's Swivel Seal or Auto-Fit hose ends and Aeroquip's Little Gem hose end with either Earl's ES 400 Perform-O-Flex or Aeroquip's 601 hose. I will not use anything else.

None of the foregoing should be taken to mean that I advocate the indiscriminate use of expensive plumbing bits. I do not. I also do not believe in settling for second best if it means a decrease in reliability—let alone safety. It does mean that I do not approve of the use of street car plumbing on racing cars. Conversely, I think that it is downright foolish to use the expensive race car/aircraft quality parts on street cars and ski boats. At intervals of two to three years, I replace the OEM flexible plumbing on my tow vehicles and personal cars. I have had one engine fire when a European OEM fuel hose turned out to be something less than adequate for extended underhood temperatures, and I am damned well not going to have another. For replacement I use Earl's Super Stock fabric reinforced hose and push-on hose ends.

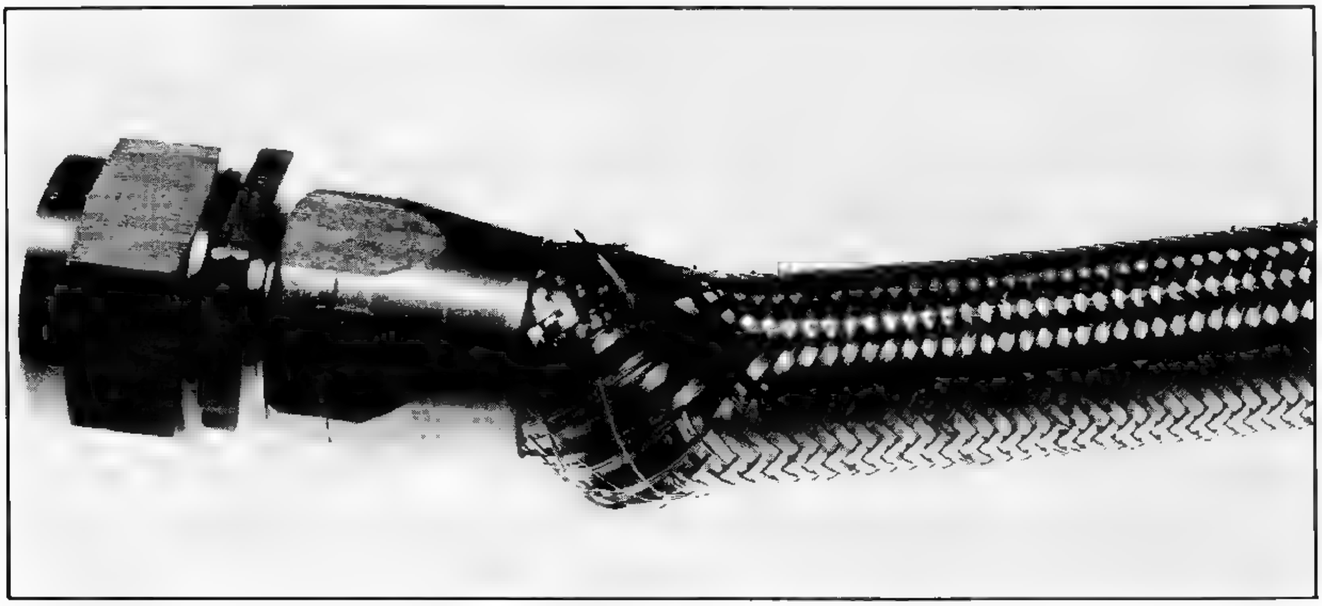
Hoses

There are at least three reputable manufacturers of stainless steel braid protected hose in this country. If there is any functional or visual difference between their products, I have not found it. This is just as well, since I cannot tell who made any particular length of good hose and I am not convinced that anyone else can either. However, quality control being what it is, some substandard stuff does slip through inspection and there have been some bad experiences.

To the best of my knowledge the only actual failures of quality hoses have been caused by the outer sheath of protective braid not getting bonded to the hose liner in manufacture. The result is a hose that can collapse when subjected to suction (as in the scavenge side of a dry-sump oil system), causing instant disaster. In this particular case the braid not only wasn't bonded to the liner, it wasn't even attached; you could pull the liner out of the



Earl's Performance Products Speedflex hose end cutaway. The inner tube is made of Teflon with the outer tube of corrosion- and fire-resistant stainless steel. Temperature range is –73°C to +232°C.



Earl's hose failure at 4800 psi in a test rig. Note that hose end stayed put.

sheath like pulling a candy bar from its wrapper. It should have been impossible for anyone to assemble a hose end onto this defective hose without realizing that something was wrong—but Murphy is always looking for an opportunity and the defect wasn't noticed.

I suppose that this is as good a time as any to discuss Smith's treatise (or diatribe?) on modern manufacturing, quality control and the responsibility of the end user in the age of mass production.

As I said, someone can always do it cheaper, and just because a hose is sheathed in shiny wire braid does not mean that it is quality hose. In this case, most of the manufacturing "someones" seem to be located in Taiwan. There are a number of metal braid protected hoses being sold, sometimes as first-class stuff, that are at least as much decorative as they are functional. I have no objection to this sort of thing so long as the end users are informed that what they are buying is not of aircraft or race car quality. This is not always the case. Some of the merchandisers are ignorant, and some are just plain dishonest.

Fortunately, there are a couple of relatively easy ways to judge the quality of lightweight fuel and oil hose. The illustration shows the three possible configurations of stainless steel braid protected synthetic rubber flexible hoses. One view shows the right stuff—in this case, Earl's ES 400 Perform-O-Flex competition hose. Note that there are actually two protective sheaths of stainless wire braid—the full coverage outer sheath that everyone knows about because it is highly visible, and the normally invisible partial coverage inner braid located between the inner liner and the outer rubber jacket. If the hose you are looking at does not have both sheaths, and if the outer sheath is not bonded to the rubber jacket, then, regardless of what the salesman is telling you, you are not looking at race car-quality hose. In an effort to provide easy visual identification, both Earl's and Aeroquip now incorporate a tracer strand of slightly different color stainless braid in the outer cover. Unfortunately this is easy to do, and while the absence of a tracer does mean that the hose is not the right stuff, the presence of the tracer does not necessarily mean that it is.

Another view shows Earl's Econ-O-Braid hose. Notice that the inner protective braid is fabric, not stainless steel wire. The outer braid is not bonded to the liner so that you would notice. This particular hose is a lot cheaper than the righteous stuff. It is perfectly suitable for street and custom cars, ski boats and such less than serious devices. In my opinion (and in Earl's) it is just not good enough for race cars.

A third view shows the moonglow hose from across the western ocean. There is no inner braid. The purpose of the outer braid is decorative. The quality of the hose as well as its pressure and temperature capacity is unknown.

An alternative on this same theme is the use of tin-plated copper braid with the same hose construction—cheaper yet, and serviceable, but not for racing.

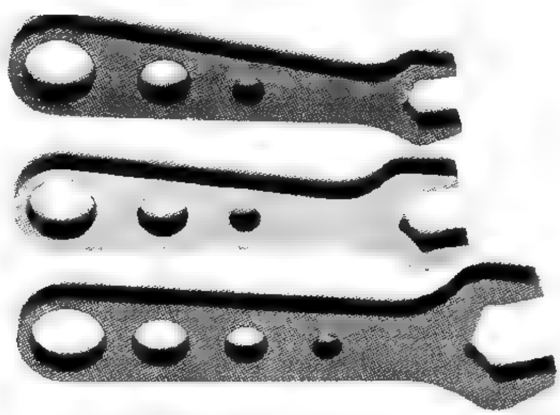
I guess it must be that I just don't like imitations. If, for whatever reason, I am not going to use the very best, then I pass on the whole metal braided protected hose bit and use Earl's Super Stock hose. Some drawbacks are that it does not offer the abrasion or flame resistance of the stainless protected hoses and it is heavier, less flexible and harder to keep clean, but it is hell for stout and has the same temperature rating as the right stuff. It is also not as stiff as the metal braid protected line, and it is possible to kink the hose if the bend radius is too tight. It is also possible for it to collapse in suction applications. Fully aware of this, Earl's recommends and supplies an inner coil spring to be inserted in the hose for those situations. It also comes in four colors. I use it on all of my street cars and tow vehicles and for vent and overflow lines on the racing cars.

How to do it

At the beginning of this chapter I stated that most current plumbing failures are the result of misuse of the right components. Again with Earl's kind permission, I am going to repeat my words in the assembly and installation of hose and hose ends from their catalog. Both the words and the illustrations also apply to Aeroquip—but not to other brands.

Mistakes—Or how to screw up the foolproof stuff

There are several common errors made during hose end assembly.



Hose end wrenches.

Smith's treatise on the end user's responsibility

Just about everything that we use—including our racing cars, our airplanes and even our space vehicles—is, to some extent, mass produced. This is a necessary evil that comes with our way of life. It is the only way that the cost of goods can be brought down to the level where the masses (us) can afford the proliferation of technical goodies that we enjoy.

It is not economically feasible to individually inspect each mass produced item before it is sold. Instead a small percentage of each production run is normally sampled. If a sample is found to be faulty the line is stopped and the defect in the manufacturing process that caused the fault is rectified. Hopefully all of the parts finished since the last satisfactory sampling will be inspected and the bad parts (if any) will be rejected, reworked or whatever.

The system works pretty damned well and, although we bitch about quality all the time, we actually purchase very few defective items. I recently heard the quality control standards of a certain industry compared to those of the human race when practicing reproduction. No matter how stringent the quality control inspections may be, some defects will always sneak through.

I believe that the end user must share the liability for the use of such parts with the manufacturer. For example, I X-ray inspect my brand new race car wheels—and I expect my engine builder(s) to X-ray inspect brand new cranks, rods, valves and so on before they are assembled into my engine. If I spend ten minutes trying to screw a nut onto a bolt only to finally discover that the nut slipped through inspection with no threads, then I must share the blame—I should have seen it (or at least felt it)! If I assemble a hose end onto a hose when the sheath is not bonded to the liner, or if I install a hose end in which the sealing taper did not get machined, a portion—even a major portion—of the fault is mine.

The liberal judges who have taken over our court system do not agree with this notion—or, for that matter, with any aspect of personal responsibility other than taxpaying. But that is not my department. The judge is not going to be of any immediate use if you install a defective part—so form the habit of visually inspecting every part before you assemble it or install it. If it doesn't feel right going together or going on there is always a reason. Take it apart (or off, as the case may be) and find out what is wrong before it strikes you dead. Once a failure has happened, assigning blame is not going to help anything or anybody.

First, many people do not cut the hose straight. This makes it impossible to engage the sharp edge of the cutter squarely with the hose inner liner and can, in extreme cases, result in the creation of the dreaded flapper, a leaking assembly or both. In addition to the normal radial wheel or fine toothed hacksaw method of cutting hose, it is also possible (and easy) to cut the stuff with a really sharp wide bladed chisel—and a really big brass or copper hammer. This is, in fact, my usual method. The tricks are:

- The hose must be backed up by something really solid—I use an old chunk of 4 in. round steel bar that I clean up on a lathe when the chisel scars get too bad.
- You have to hit the thing really hard—one shot is all that you get.
- The hose must be very tightly taped. I prefer masking tape to duct tape.
- This method is definitely not suitable for Teflon hose—it crushes the Teflon.

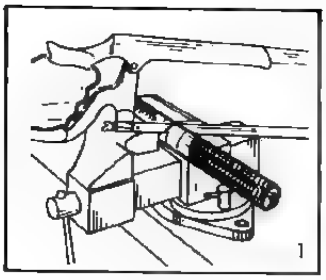
Many people allow the outer protective metal braid to fray and/or partially unravel when cutting the hose. This can be a result either of not taping tight enough or of the use of a coarse toothed or dull hacksaw blade. The unravelling makes it difficult (in extreme cases it makes it impossible) to fit the socket over the hose at the beginning of the assembly procedure. It also usually results in a multitude of tiny puncture wounds on the hands of the operator as well as the addition of small quantities of blood (a poor lubricant) to the assembly.

Many people fail to use a lubricant on the cutter—making it more difficult than it should be to start the annular separation of the inner hose liner.

Many people fail to use an anti-seize compound on the cutter threads. This can lead to the unintentional creation of a permanently assembled hose end by galling the male and female threads together forever. With a lot of bad luck it can also result in a complete inability to finish assembling the part.

Many people fail to tighten the socket sufficiently onto the nipple. This results in a lesser amount of hose retention than was designed and can lead to either leakage or, in really extreme cases, blow off. The maximum allowable gap between the faces of the assembled nipple and cutter is 0.046 in. Everyone, including me, eyeballs this dimension.

Many fail to detect the problem when the hose tries to work its way backward through the socket as the cutter advances onto the socket during assembly. This is a particularly nasty happening because the completed assembly may look perfectly normal when, in fact, the hose is neither properly retained nor properly sealed—a disaster looking for a place to happen. Murphy's laws assure us that the disaster will happen at the worst possible time—like while leading Indianapolis by

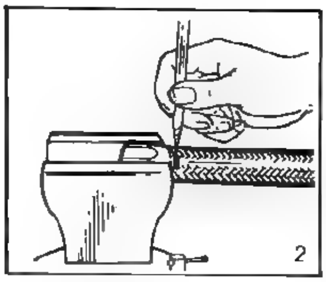


1. Cut hose to required length.

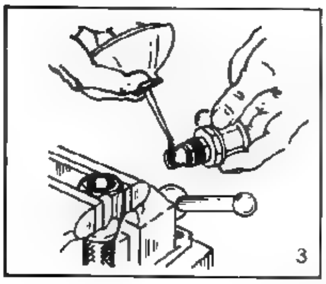
a Measure distance between ports or adapter fittings along the path that the hose run will follow allowing for bend radius, hose end length and offset to obtain length and hose required.

b Cut the hose square with a radiac wheel or a sharp 32 teeth per inch hacksaw blade. It is necessary to wrap it tightly with masking tape before cutting and to cut through the tape. This helps to prevent the stainless wire braid from fraying.

c Trim any frayed end of the braid with a sharp pair of metal snips or diagonal cutters and remove the tape.

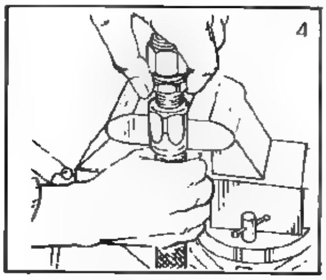


2. Place the socket in a vise and insert the end of the hose into the socket until the hose butts against the bottom of the threads provided for the cutter. Gently pull the hose back until there is a 1/16" to 1/8" gap between the end of the hose and the bottom of the socket mark hose at bottom of socket with a felt pen so that you can detect any tendency of the hose to be pushed out as you complete the assembly.



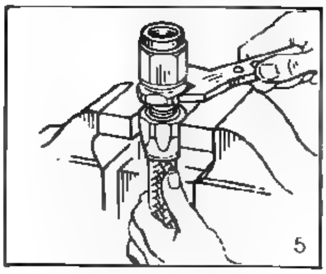
3. Lubricate the inside of the hose, the cutter threads and the socket threads. Just about any kind of clean oil will do but I prefer to use an anti-seize compound on the threads. Place the nipple in a vise. Flex muscles.

4. Push the hose and the socket onto the nipple until the socket threads can be started on the cutter. Start the threads and go as far as you can by hand. Depending on the size of the hose, some force may be necessary in this part of the operation.



5. To complete the assembly it doesn't matter whether the nipple or the socket is held in the vise. Holding one or the other in the vise and using a suitable wrench on the other, tighten the socket onto the cutter threads until the socket is within .060" of bottoming on the nipple. Do not use an adjustable or over-size wrench or you will damage either the nipple or the socket.

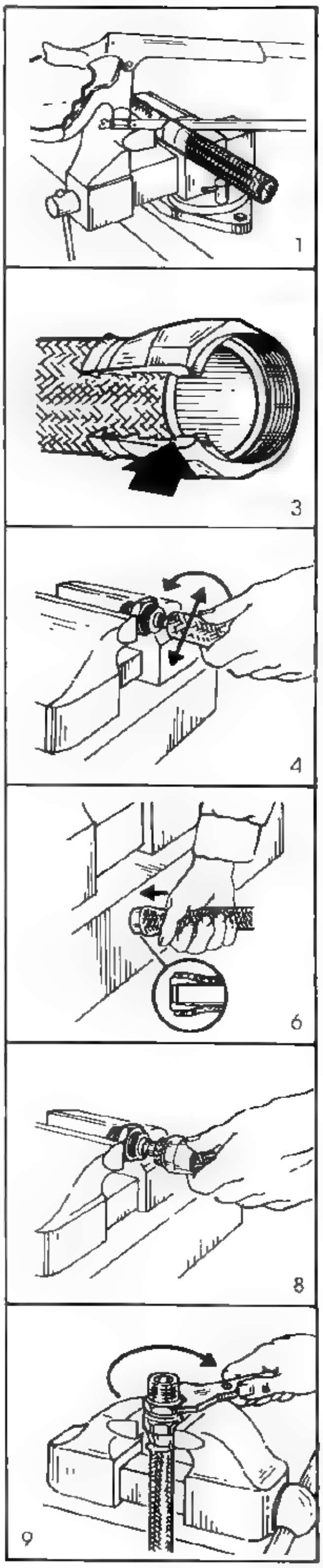
6. Check the mark that you made on the hose in step 2. If the hose has backed more than about 1/16" out of the socket as you assembled it curse and return to step 3.



7. Clean the hose and the hose ends with CLEAN solvent.

8. It is most unlikely that you will have available any method of pressure checking the assembly before it is installed. Before letting the assembly out of your sight check the assembly by running the system at full pressure while you observe the hose, hose ends and adapters for leaks.

Assembly tips for Earl's Perform-O-Flex hose. Earl's



Brake lines are critical items. The potential penalties for improper assembly are severe. Although there is nothing complicated about the procedure and no special tools are required, extreme care must be used in assembly. We strongly recommend that the following procedures be used.

1. Cut the hose to the required length. We recommend the use of a radiac wheel but it can be done satisfactorily with a 32 teeth per inch hacksaw blade. In either case, the hose must be tightly wrapped with masking tape and the cut made through the tape. Do not cut FLUOR-O-FLEX hose with a chisel, snips, pliers, or a shear as these may crush the Teflon sleeve.

2. Deburr the Teflon and trim any loose ends of braid with sharp snips or diagonal cutting pliers.

3. Install the socket on the hose with the threaded end of the socket toward the cut end of the hose. This will be a lot easier and you will end up with fewer holes in your hand if you clamp the socket in a vise. Push socket on well beyond end

4. Place the hex portion of the nipple in the vise. Insert the end of the hose onto the nipple and bottom the hose against the chamfer seat of the nipple with a rotary motion of the hose. This will size the I.D. of the Teflon tube and start the necessary separation of the tube from the braid.

5. Separate the braid from the O.D. of the Teflon Tube with a small screwdriver or a scribe. Be careful not to scratch or nick the Teflon.

6. Install the sleeve between the braid and the Teflon tube. Make sure that none of the braid is trapped between the Teflon and the sleeve. Bottom the tube against the shoulder of the sleeve and make sure that the sleeve is inserted square.

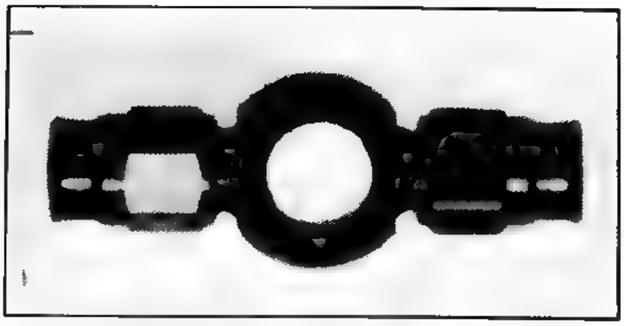
7. With the nipple held in the vise, push the hose and the sleeve onto the nipple until the sleeve bottoms. Remove the hose and make sure that the Teflon tube is still bottomed against the shoulder of the sleeve and that the sleeve is still square.

8. Push the hose and sleeve back onto the nipple and bottom against the chamfer. Start the socket onto the nipple threads and hand tighten.

9. Place the socket in the vise and complete the assembly by tightening the nipple onto the socket with a wrench until the gap between the face of the socket and the hex of the nipple is .023" to .046" – use a feeler gauge.

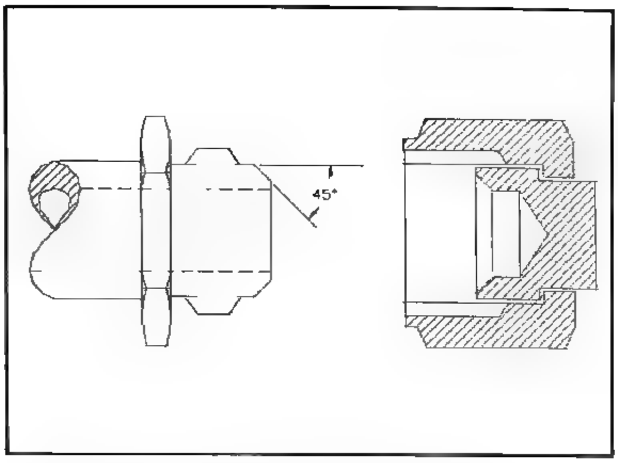
10. Blow the assembly clean and pressure test before running the car.

Assembly tips for Earl's Speed-Seal and hose ends.
Earl's

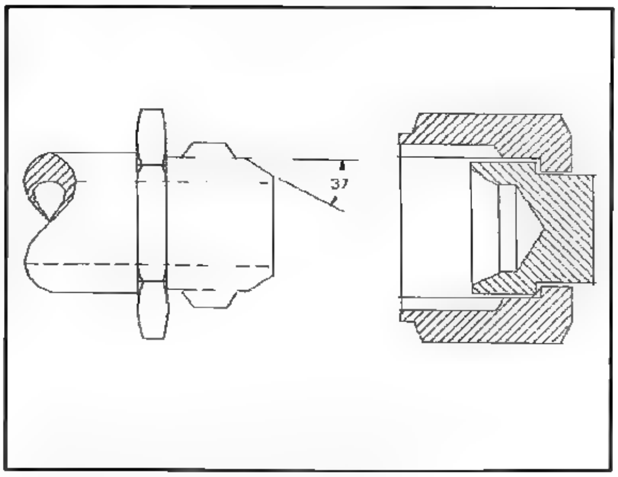


Earl's Performance Products hose end tee.

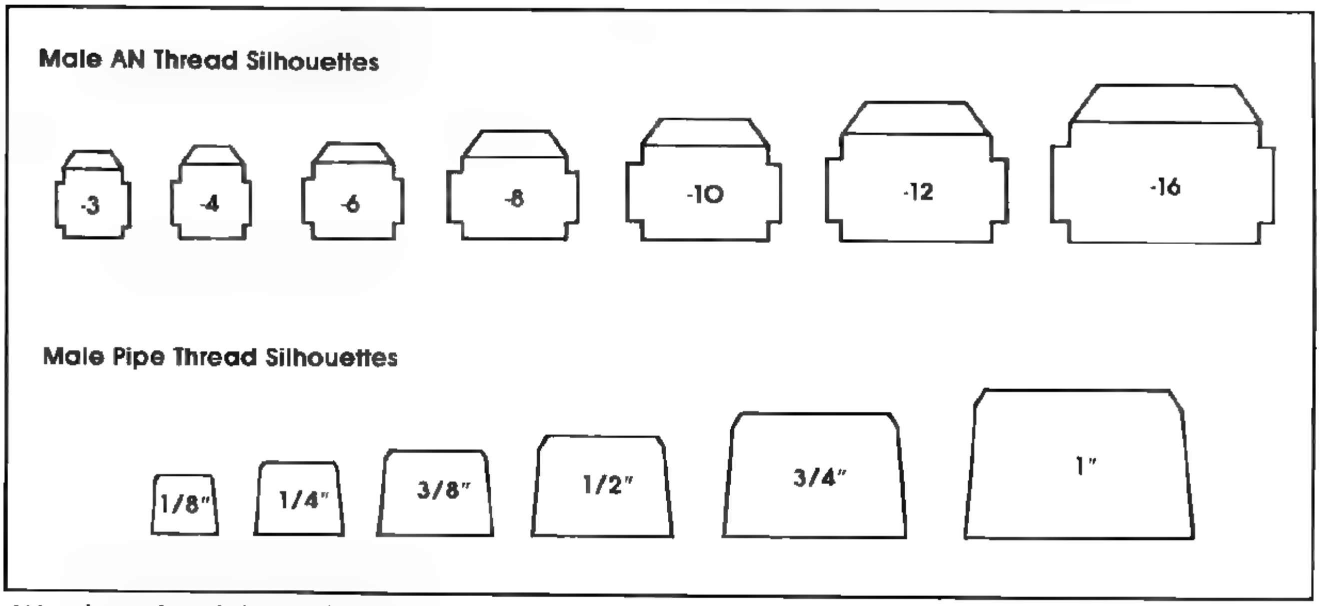
Trick homemade Y adaptor from Torino Motor Racing.
Roy Kiesling



A 45 degree male JIC cone and female flare.



A 37 degree AN sealing cone and flare.



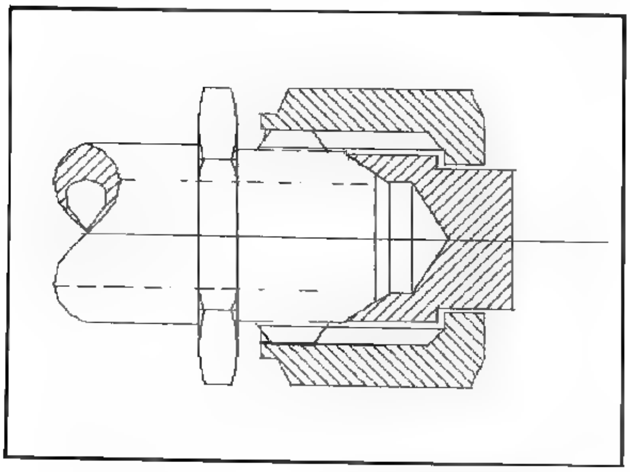
AN and NPT thread chart and silhouettes.

miles with twenty-five laps to go (yes, it did happen and no, I'm not going to tell who it was). The only way to positively prevent this sort of embarrassment is to mark the hose as described in assembly step two and to check the mark after assembly—every time.

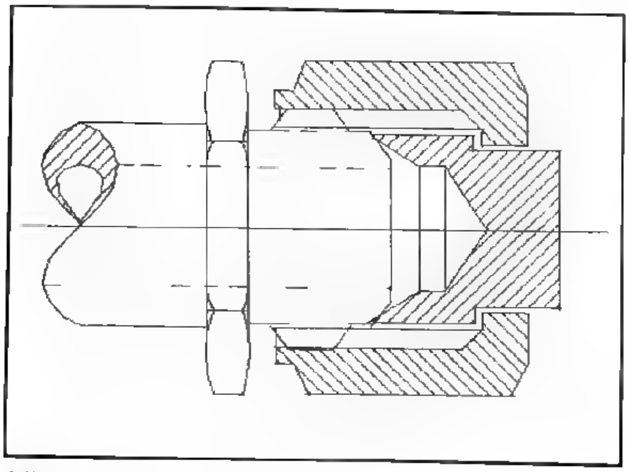
Beware attempting to replace a hose end without re-cutting the end of the hose. It is miracle enough that the nipple cutter does its thing the first time. Asking it to do it again in the old wound is a bit much—the odds of success are not in your favor. These odds, in fact, strongly favor the creation of a flapper, a blown off hose end or a giant leak. The same is true of the single-nipple hose end. Cut the hose back to virgin rubber and start over.

Do not attempt to assemble a previously used nipple and cutter hose end with a ring of inner liner rubber trapped in the annular sealing chamber. As you would expect, this just doesn't work at all.

The aircraft hose ends and adaptor fittings are designed to AN and MS specs which specify a 37 degree cone angle. Industrial hose ends and adaptors are designed to JIC (Joint Industrial Council) specs with a 45 degree flare angle. The stuff that you buy at the auto parts or industrial supply store will be JIC, and the stuff that you buy at the airplane or race parts store will be AN. The two will not mix. Be careful.



A 37 degree AN sealing cone and flare assembled.



Mismatched AN and JIC flares.

Reuse of hose ends

All single-nipple, double-nipple and push-on hose ends are completely reuseable, as is the hose—once you have removed the extreme ends of the hose. When disassembling a double-nipple hose end, it is not at all uncommon for the inner tube of the hose liner (the part that is captured between the nipple and the cutter) to be torn off by the friction between the rubber and the walls of the nipple, and to remain trapped in place. When this happens the ring of rubber must be removed before the nipple can be used again—and it is a bear to get out.

The disassembly procedure for the Swivel Seal angled hose ends is slightly different from that for the nonadjustable ones. With the Swivel Seal, place the socket in a soft jawed vise and, with one wrench on the nipple and another on the cutter, hold the nipple and turn the cutter until the socket is disengaged completely, then pull the hose off the nipple. This method will work only with the Swivel Seals.

With the non-Swivel Seal hose ends, you hold the socket in the vise and turn the nipple. The chances of leaving a rubber ring are increased simply because the rubber under shearing stress has two moving surfaces to deal with rather than one.

If you should be so unfortunate as to end up with a ring of rubber trapped between the nipple tube and the cutter I.D., do not despair. Grind one end of an old hacksaw blade into a flat hook. Insert the hook into the chamber containing the recalcitrant rubber and pull it out.

Super-light hose

In most international classes of racing it is no great task to get a car down to its minimum weight, which is defined by reasonable regulations. The same is not true, however, of Formula One cars. Since Formula One is the leading edge of automotive technology, the weight limit has been set low—the top of the heap is never meant to be easy. Since any designer and any driver will cheerfully sell their sisters for a couple ounces of car weight, it is not surprising that Formula One has given birth to a new family of extremely lightweight flexible fluid hoses designated Speedflex II.

The inner liner of this family is formed of thin-wall convoluted Teflon. A spiral protective sheath of woven Nomex and fiberglass is bonded to the

Teflon, and the assembly is protected by a woven braid of stainless wire or, in the case of Speedflex II, SL Nomex, or Kevlar/Nomex. The hose is truly featherweight. It is not as strong, as damage resistant or as abrasion resistant as the familiar stainless braid protected hose and, as you would expect, it is brutally expensive.

It also requires yet another type of hose end. With the Speed Seal II hose end, the protective outer braid is skinned back and the Teflon hose liner with its bonded layer of Nomex is threaded through the aluminum ferrule. The Teflon/Nomex hose is then cut off flush and square with the end of the ferrule, and the outer braid is trimmed square and pulled up to about ⅛ in. back from the edge of the ferrule. The socket is then pulled forward over the ferrule and the braid, and the hose end is screwed into the socket. The Teflon convolutions are progressively jammed together and wedged between the nipple and the olive. The assembly is sort of a cross between Earl's Auto-Fit, and Earl's Speedflex and Aeroquip's Super Gem. The hose end and socket are reuseable, as is the hose after the ends are trimmed off. The soft aluminum ferrule is not reuseable, in fact, it is not even removable.

Pressure testing

While Earl's and Aeroquip have made the field assembly of stainless protected braided hoses practical—and even simple—they have not made it foolproof. For the quarter century that I have been assembling hoses, I have been worrying whether or not the damned things are going to leak when I put them on the car. To be honest, I never have had one actually leak, but I still worry. A couple of months ago I actually did something about it. I made up a hose test kit. Why I didn't do it twenty years ago, I do not know.

I turned the base of a standard all-metal tire valve down to slightly smaller than the I.D. of a dash twelve AN plug, drilled a ⅜ in. hole through the plug and installed the valve (with its rubber gasket) in the plug with the filler facing away from the threads. The hoses that I wanted to test were dash twelve, so I merely plugged one end of the hose, installed the tire valve plug in the other end and pressurized the assembly. I used a Fox/Penske shock charging valve to do the testing because it has a pressure gauge built in, but a standard high-pressure tire gauge will do just as well. Leaks are detected either on the gauge or by spraying the hose end assembly with 409 all-purpose cleaner.

I completed the kit by making up a short length of dash twelve hose and a group of male-to-male AN adaptors from dash twelve to every size that I use.

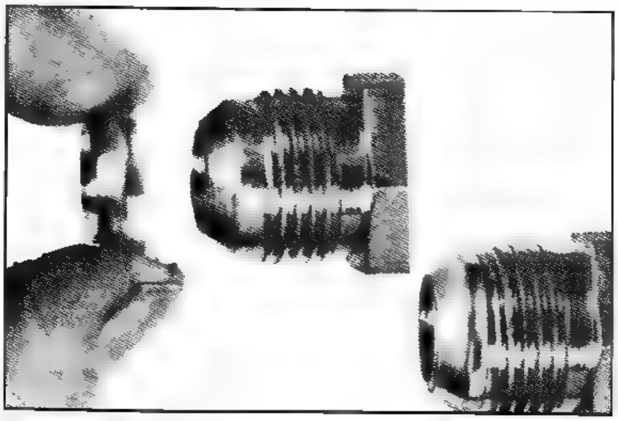
Leaks

If a quality nipple style hose end fitting leaks it has either been assembled incorrectly, or the sealing surfaces on the adaptor and/or the nipple have been damaged. Or, just possibly, someone has tried to assemble a 37 degree AN seat onto a 45 degree SAE cone. Damage to the cone or the seat can be caused by a multitude of sins—dirt and overtightening being the most common. We are not about to do away with either one. The good news is that we now have an instant fix for the damaged sealing surface—the conical seal.

What the hell is a conical seal? Just about the neatest thing to come out of California since the founding of the wine industry! Have a look at the photos shown here.

The conical seal is formed from a malleable alloy (either aluminum or copper) that flows into any imperfections in the sealing surfaces. The metal is malleable enough to form a leakproof joint with standard assembly pressures. It is installed by simply pushing it over the end of the male cone and assembling the hose end in the normal way. Equally spaced friction flats on the conical seal retain the device on the male cone and prevent misalignment. I have used them about eight times in the past six years. One of those times, a dash three size conical seal won a championship race for us. I carry a full range of the things in my toolbox and consider anyone who knows about them and does not carry them to be foolish. They are formed for the 37 degree AN cone but are malleable enough so that they will work with 45 degree JIC cones.

There is one other common cause of hose end leaks—people don't tighten them. The only way I know to be certain that every hose end is properly tightened is for everyone concerned to form the habit of never leaving an adaptor, a hose end (or, for that matter, anything else) loose, finger-tight or partially tightened. Even when you know that you are going to take the thing off again in two minutes, properly tighten it—*every time.* Otherwise Murphy will eventually get you.

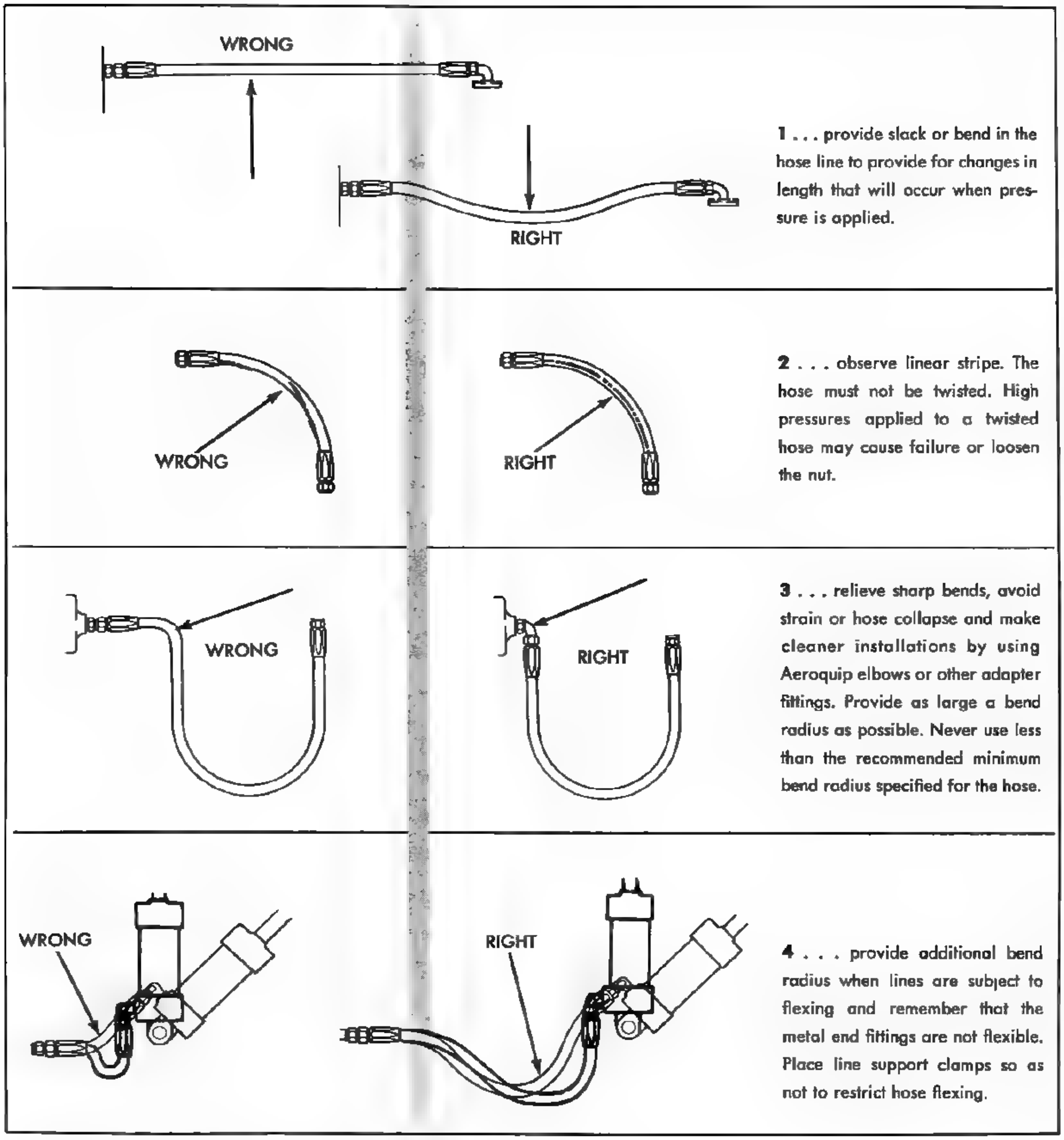


The conical seal.

To finish our discussion of leaks, I will state one more time that there is no need whatsoever to put Teflon tape, gasket goop or anything else on either the sealing surfaces or the threads of the coupler or the adaptor—it only makes a mess.

Hydraulic systems

More than twenty years ago we figured out that the swelling of conventional flexible brake lines under pressure was using up more of the available brake pedal travel than was either wise or



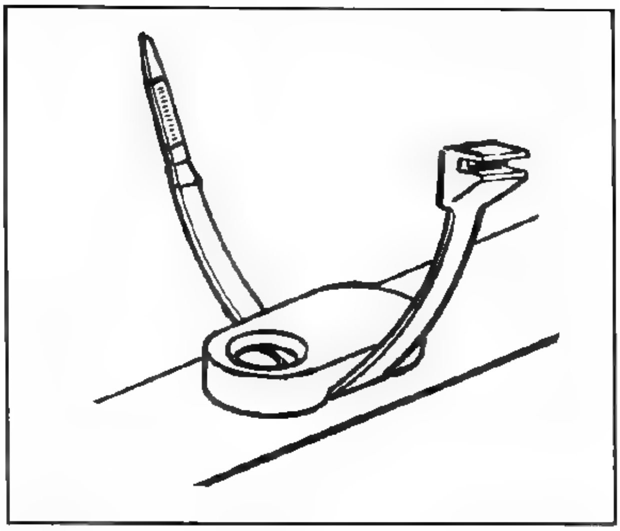
Right and wrong ways to run hose. Aeroquip Corporation

necessary. We solved the problem by calling an Aeroquip service engineer. Acting on his advice, we changed over to a Teflon hose with a stainless steel braid protective cover and reuseable hose ends which had been developed by Aeroquip for use in aircraft and missile fluid systems.

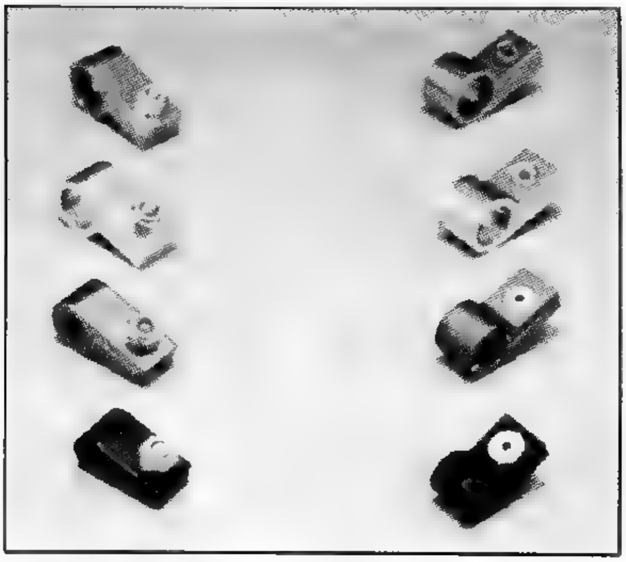
Virtually nothing else is currently used on racing car brake and clutch systems. There are a number of aftermarket kits for both automotive and motorcycle brake systems. This is one of the areas where we have to be really careful. As always, all that glitters is not gold and there are pitfalls in store for the unwary and the uninformed. Since what we are talking about here is possible sudden failure of the braking system, these particular pits tend to be particularly deep and nasty when you fall in.

As you would expect, there are exactly two manufacturers whose hose and hose ends I will allow on the braking systems of my racing cars. To no one's surprise, they are Earl's Performance Products and the Aeroquip Corporation. The quality and the function of the two product lines is identical. Earl's offers a greater variety of hose end configurations—including the hose end tee pictured here, which I now use (instead of the usual AN tee adaptor and three separate hose ends) at the rear brake line junction. At the front I use a double banjo directly at the master cylinder. I have found that a great many racers do not understand the inner workings of these hose ends either, so stand by for one more explanation. Refer to the cut-away shown—an Earl's Speed Seal which is functionally identical to the Aeroquip Super Gem.

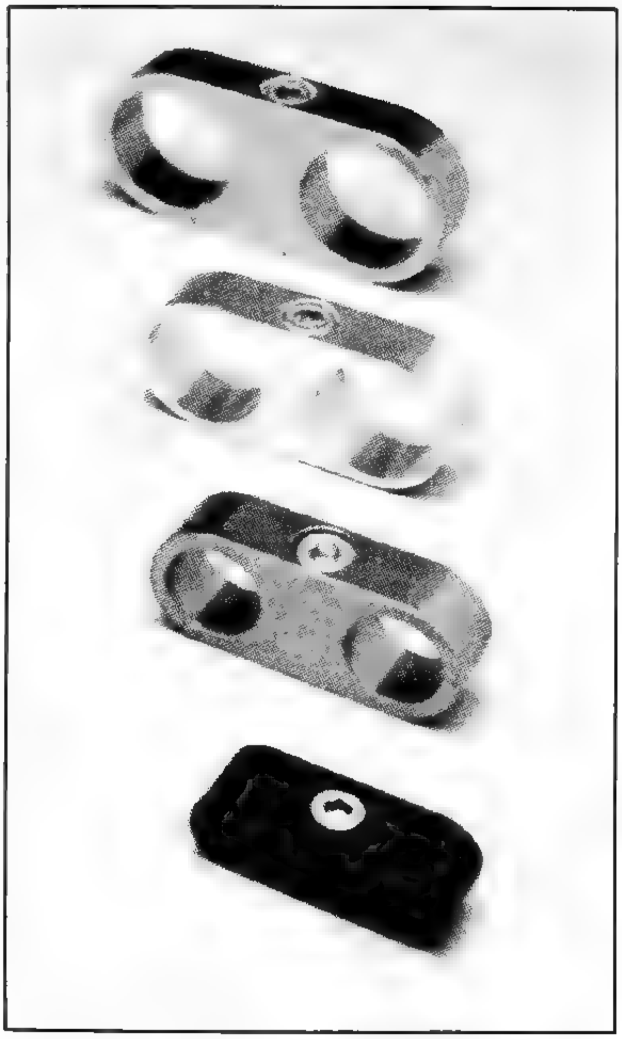
The sleeve is inserted between the Teflon tube and the stainless steel protective braid before assembly. When the nipple is screwed into the



Ti-wrap and ti-wrap saddle.



Earl's hose clamps.



Earl's hose separators.

socket, hose retention is ensured by the wedging of the braid between the I.D. of the socket—which is shaped and sized to produce a predetermined amount of crush—and the O.D. of the sleeve. At the same time, the action of the tapered I.D. of the socket against the O.D. of the somewhat flexible sleeve forces the annular barbs on the I.D. of the sleeve into the Teflon tube and butts the taper at the face of the sleeve against the seat on the nipple. This forms a two-stage seal. The sealing function is separated from the hose retention function and the strongest possible assembly is obtained. The sleeves are available in either stainless steel or brass. I prefer the brass because it is softer and should seal better, but I have never had any trouble with the stainless.

Assembly

Once more, this comes straight from Earl's catalog:

Doing It Right

There are not a whole lot of no-nos in the actual installation of flexible plumbing runs. About the only things that can be done really wrong are: first, to install a hose under tension, either axial or radial (and *that* will be pretty obvious when you go to hook up the hose). And second, to install a hose in such a way that it will interfere with something—or be interfered with—under some combination of dynamic conditions.

It is not unusual for a really neat and convenient hose location to suddenly become all wrong when the suspension travels, or the front wheels turn, or the fluids involved get hot and burn the driver (or heat the fuel), or the exhaust gets hot and boils the brake fluid, or the car gets off the road and tears off whatever hoses were dangling—even a little bit. This last happening will leave the driver with no—select one or more of the following: brakes, clutch, fuel delivery, oil and/or water—but with a fearsome temper. The specific no-nos that I want to warn you about include: first, leaving insufficient clearance between each hose end and anything that it might be able to contact or vibrate against. While the hose is flexible, the hose ends are not.

Second, allowing a hose to come in contact with a sharp corner, a nut, a bolt, a rivet stem or anything else that is not perfectly smooth. This one includes failure to install a grommet at each point that a hose passes through a panel.

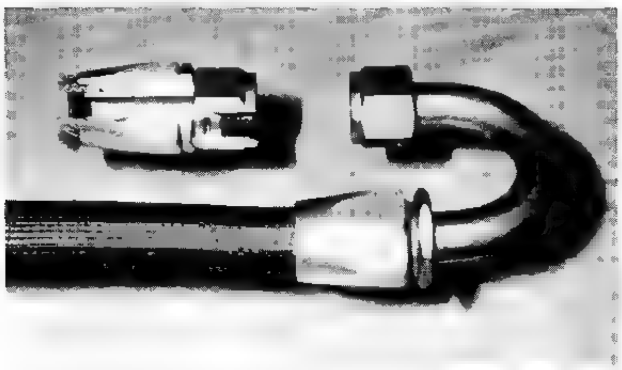
Third, allowing a hose to rub against *anything*—even when the surface against which it will rub is flat and smooth. The stainless braid makes a very efficient file and will abrade through anything that it moves against. This is particularly true in those instances where brake and clutch lines pass through the fuel cell compartment. In this case I encase the hoses in a thin walled aluminum tube. Spiral wrap is a neat and convenient way to prevent chafe damage under normal conditions.

Fourth, kinking the hose—either by bending it

Minimum bend radii for aluminum hard lines 1100–H14, 3003–H14 or 5052–0

Tube OD (in.)	Min. bend radius (in.)
1/8	3/8
3/16	7/16
1/4	9/16
5/16	3/4
3/8	15/16
1/2	1 1/4
5/8	1 1/2
3/4	1 3/4
1	[illegible]
1 1/4	3 3/4
1 1/2	5
1 3/4	7
2	8

Minimum bend radii for aluminum hard lines: 1100–H14, 3003–H14 or 5052–0.



Earl's Superlight Speedflex hose and Speed Seal hose end.

It is possible to run hose ends as well as hoses too close to something: here is a damaged end.

too tightly (both Earl's and Aeroquip include minimum bend radii tables in their catalog) or by placing the hose in a torsional bind.

Fifth, overtightening the hose ends onto their adaptor fittings or into their ports. Both the seal and the self-locking feature are provided by the design, not by force. It helps a lot to use the wrenches made for the job. Their handles are short enough to make overtightening difficult.

Sixth, stretching the hose or not allowing enough room for flex. There is a right way and a wrong way to run hoses. The right way mainly calls for common sense. Some people don't have very much of it.

Finally, don't allow things to hang by their hoses. This is particularly true of brake calipers. It is the single most common cause of failure in brake system hose ends, and is the main reason that I do not allow the use of 90 degree hose ends at the caliper. What happens is that the hose end gets bent at its weakest point. The bend is a stress raiser and, sometime later, the hose end fails from fatigue. Not pleasant at all! Since I don't even trust myself much—let alone anyone else, I use Earl's banjo hose end on all of my calipers.

Maintenance

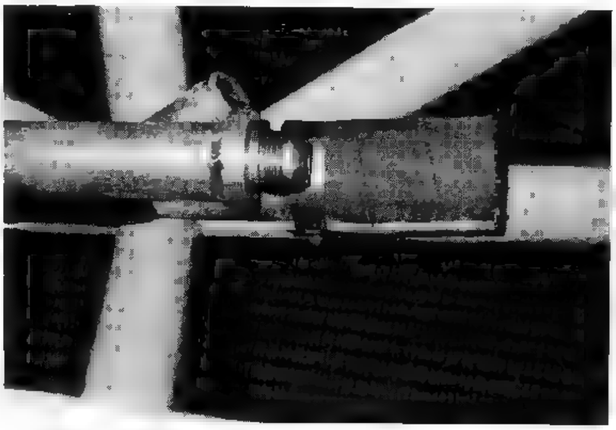
Plumbing systems require virtually no maintenance at all. The maintenance that is required is largely a question of preventing abuse. There are several ways to help avoid plumbing problems.

Inspect the whole system frequently for signs of chafing, abrasion, crushing or seepage. This is one of the things that we typically don't do very well. The most effective checks are visual and tactile. Take off the hose, clean it and look at it. If the outer braid is deteriorated to any serious extent, throw the hose away—saving the hose ends. Next, take the hose in both hands and bend it back and forth. If you hear (or feel) a crackle, the hose liner has gone brittle and again you get to throw it away. Last, while bending the hose, make sure that the outer protective braid is still firmly attached to the hose liner. If not, trash it. With the Teflon lined hoses, any abrasion that has visibly worn any amount of wire is cause for rejection as is any visible kink in the hose. Rounding out a kink with a pair of pliers will only increase the damage already done to the Teflon liner. You haven't lived until you have experienced brake line failure.

Keep both the hose and the fittings clean. Before removing any hose end from its adaptor or port, wash down the assembly with solvent—or even with gasoline—and blow it both clean and dry so that no grit or dirt can find its way into the threads or sealing surfaces. As soon as the hose has been removed, install a clean protective plug into the hose end and a clean cap over the adaptor. Always inspect both the hose end and the adaptor for damage or dirt before reassembly. Race cars, particularly open-wheeled race cars, are forever getting corners either knocked off or folded back against the chassis. In either case, the flexible brake lines and their hose ends are going to be stretched and distorted. Scrap them—it just isn't worth the chance.

The only way to end up with a truly neat, serviceable and workmanlike plumbing installation is to think it all out ahead of time. He who grabs a couple of coils of hose and a shoebox full of hose ends and adaptors and gets stuck into it will inevitably end up with an unmanageable, unpresentable and expensive bunch of silver worms. Planning is a question of working out what has to run where, what line size is required and then deciding on the routing and grouping of the hoses and configuration of hose ends and adaptors that will result in the neatest, most maintainable and economical installation. It helps to work out a schematic diagram listing the length of hoses, the styles of hose ends

AN-818 and AN 819 coupling nut and compression sleeve used to adapt brake hard line to AN adaptor.

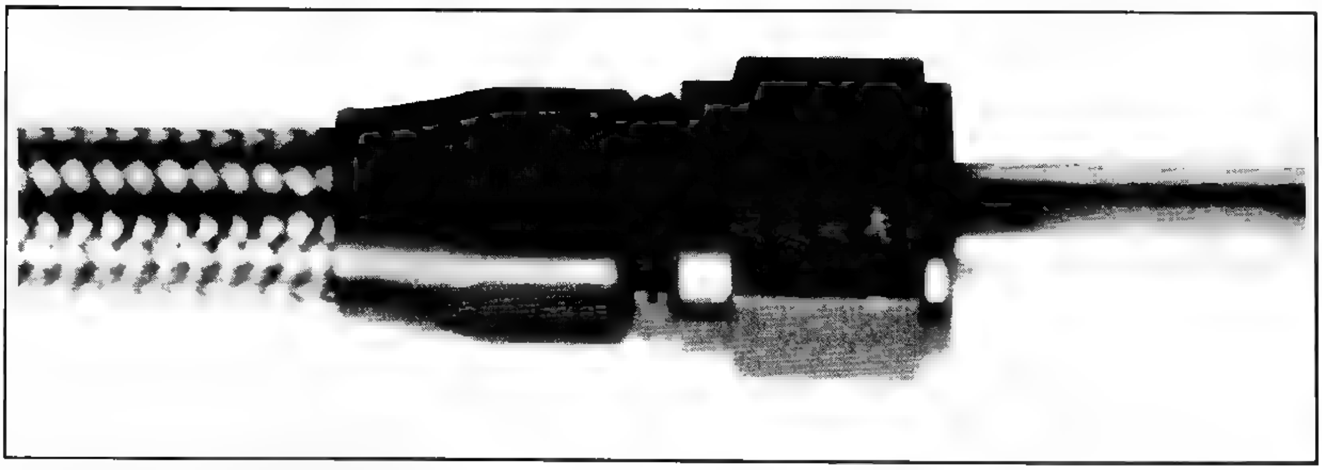


Hard line with welded AN adaptor fitting.

and the configuration and sizes of adaptors that will be needed.

When laying out the plumbing, try to keep normal maintenance functions and operations in mind. Too many good looking installations end up causing needless work—like deplumbing the transaxle oil cooler to change gears, or having to disconnect oil hoses to change the filter, or having to bleed the clutch and brakes every time that the engine is changed. You should be vitally concerned with arranging the minimum number of connections to be undone for an engine change and, even more important, the absolute minimum possibility of damaging things during engine changes. Improbable as it may seem, this happens all the time because hoses that had to be disconnected are left dangling where they can be run over by the cherry picker or where they can be crushed between either the incoming or the outgoing engine and chassis or whatever. Of course, if it doesn't have to be disconnected, it is not going to get dirt in it—and it will not have to be either reconnected or bled.

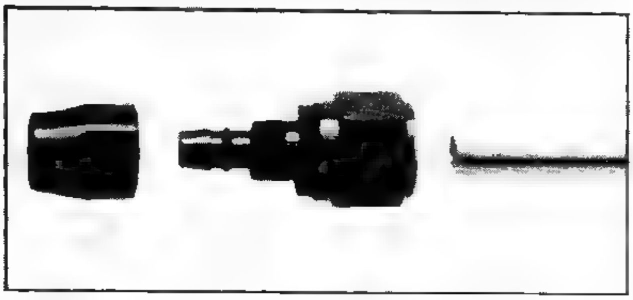
It always looks better if the plumbing lines are arranged to run together in logical groups of about the same temperature, among other things, and it also makes it a lot easier to support the hoses. The



Earl's Tube-Mate hard line adaptor.

TUBE-MATE

Now connecting to steel, copper and aluminum tubing or beaded or barbed connections, is easier, faster and more efficient thanks to Earl's new Tube-Mate hose ends. Utilizing the proven Auto-Fit hose end, Earl's has again simplified the use of stainless steel braided hose for traditional fuel, water and oil lines with the introduction of this new series of hose ends.



INSTALLATION

For installation, simply slip the tube inside the Tube-Mate, and tighten the nut. The unique interior design piece tightens around the tube, creating a virtually leak and seal proof junction.

Part No.	Size	
	Flareless Tube-Mate	
360104	-4 hose to 1/4"	tubing
360165	-6 hose to 5/16"	tubing
360106	-6 hose to 3/8"	tubing
360108	-8 hose to 1/2"	tubing
360110	-10 hose to 5/8"	tubing
360112	-12 hose to 3/4"	tubing
360116	-16 hose to 1"	tubing

Assembly tips for Earl's Tube-Mate adaptor.

hose, flexible or not, is meant to carry fluid, not to hang something from. They are not even meant to be self-supporting. In fact, they need a good deal of outside support—at about 18 in. intervals. I use lots of tie-wrap saddles and tie wraps—they are both cheap and fast. Lately I have been using Earl's anodized aluminum saddle clamps in those applications where weight is not critical, simply because they look so good.

Hard lines: Hydraulic

I use aircraft specification flexible hydraulic lines where I need the flexibility. In many applications, particularly those that feature power-assisted brakes, I prefer to use hard lines where flexibility is not required. I often have to arrange junctions between conventional hydraulic tubing and AN adaptors.

The way to do this is to use the AN-818 compression sleeve and matching AN-819 coupling nut. This means that you must flare the end of the hard line to accept the compression sleeve. This is another area where you can get into trouble. The standard American double flare is an SAE 45 degree flare and is not suitable for use with the AN fittings which are 37 degree cones. I have spoken with a number of experienced and even respected mechanics who, while they are aware of this difference, ignore it. These people feel that, by graunching down hard enough on the coupling nut, they can seal the assembly. These people are fools! The best that can be hoped for in this case is a very narrow line seal, a slight seepage and a less than positive assembly lock. The worst is stray pieces of metal in the fluid system. I use an AN single flare which is easier to form anyway. I form the flare with a Rol-Aire 37 degree flaring tool from Aircraft Spruce.

While on the subject of hard lines for hydraulic fluid, use only genuine Bundyweld steel hydraulic tubing. Bundyweld comes either terne (tin) plated or copper plated. Under no circumstances can you ever use copper tubing for hydraulic applications—it will work harden and crack.

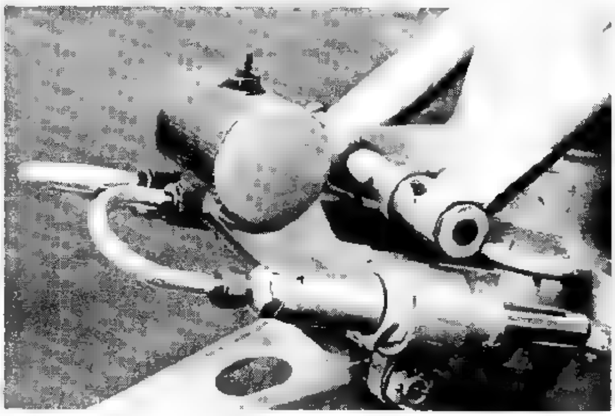
The correct sizes of hard and soft lines are 3/16 in. outside diameter hard line, AN dash three flexible line for brakes and 1/4 in. outside diameter and AN dash four for the clutch. The reason for this apparent anomaly is that the brake system transmits pressure with only minimal fluid flow, while the clutch system displaces a considerable volume of fluid. We are aiming for speed of displacement in the clutch system, so we use the larger hose. In the brake system we are looking for minimum line swelling, so it pays to use the hose with less internal surface area.

Hard lines: Fuel and lube oil

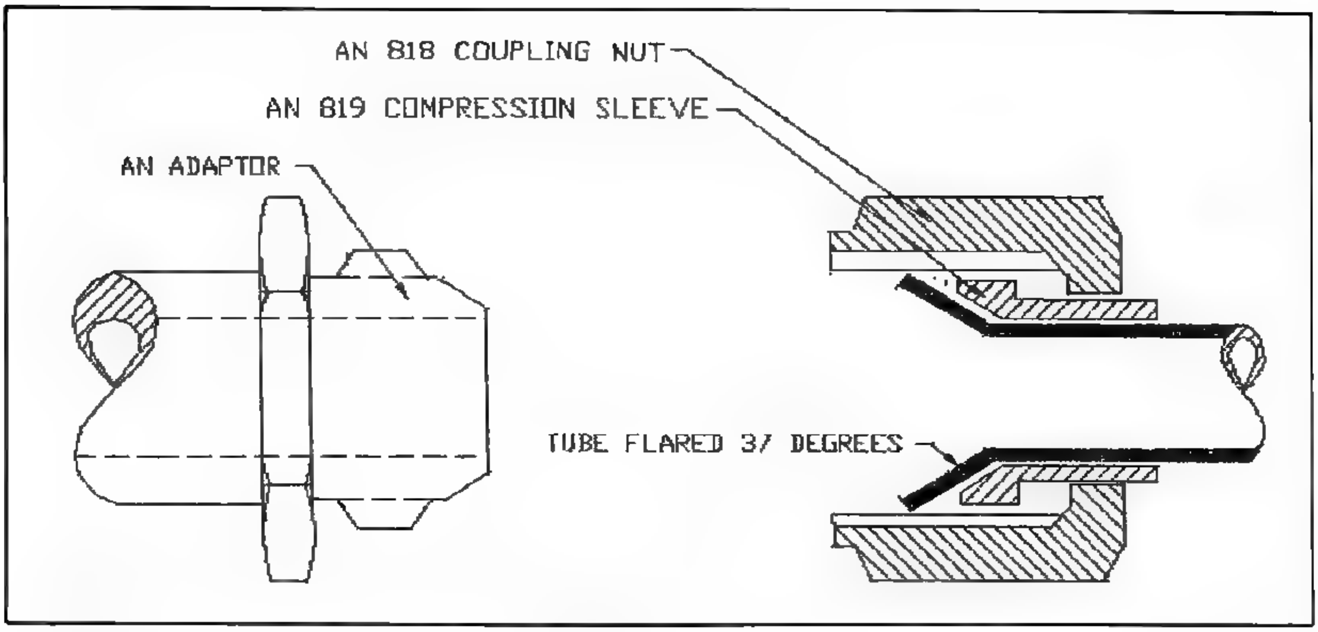
Transmission of fluids by hard lines (rigid tubing) is popular in high-performance and lightweight aircraft. Hard lines look really neat, are lighter than any hose system other than the Kevlar-wrapped corrugated Teflon ultra-expensive systems and the materials and fittings required are very inexpensive. They are a giant pain in the butt to fabricate, require expensive tooling (tubing benders and large flaring tools), leave you no margin for error and, in racing cars at least, are prone to being damaged during engine changes and the like. This is a lot less likely to happen in aircraft simply because engine changes are less frequent by a factor of several hundred and less hurried. Hard lines also require some sort of flexible connector at each end or they will vibrate apart.

I don't use hard lines for fuel at all. Their very lack of flexibility makes them susceptible to leakage due to vibration, damage and so on, and I have an aversion to fire.

Flared hard line, AN compression sleeve, AN coupling nut and AN adaptor fitting.



Flared hard line, AN compression sleeve, AN coupling nut and AN adaptor fitting used as hydraulic steering lines on P-38 Lightning airplane. Roy Kiesling



Hard lines flared 37 degrees to accept an AN–818 coupling nut and AN–819 compression sleeve.

I use hard lines for oil and breather systems when I need to save money and/or the last few units of weight. I also use them when the required bend radius is too tight for flex lines.

When it comes to the metal tubing, the basic choice is between aluminum and stainless steel. Don't even *think* about copper—it work hardens and will eventually crack. I use aluminum because it is easier to work, lighter and plenty strong enough for normal racing car use. I prefer alloy 5052–0 in 0.035 in. or 0.044 in. wall thickness for almost all of my hard lines—if it's good enough for the FAA, it's good enough for me. I bend it with a rigid tubing bender of the right size (borrowed or rented) to the bend radii shown in the illustration.

Until recently, our choices for end fittings on hard lines were limited to two. We could weld a male AN fitting of the appropriate size onto the end of the tube (always leaving the hex on the AN fitting so that we can get a wrench on the thing). Or we could form a 37-degree flare on the end of the tubing and use the AN–818 coupling nut and AN–819 tubing sleeve. Aluminum tubing should be double flared in diameters up to and including ⅜ in., and single flared in the larger diameters; double flares are not necessary in stainless tubing. All flares must be carefully formed, concentric, regular, smooth and free of nicks and burrs. It helps to cut the tubing square and to completely deburr it before flaring. Both of these options are still available and valid.

There is now a third, and considerably more convenient, option. Earl's now offers their Tube Mate hose end which attaches a standard Earl's Swivel Seal hose end to a length of metal tubing without flaring the tube or welding. The Tube Mate is available in sizes dash six through dash sixteen in straight configurations only. Sometimes life does get a little easier.

Summary

It seems to me that this has been a long chapter. I hope that I have not rambled on too much, but I felt that a detailed treatment was in order, due to the number of ignorance-type failures I have seen in the past few years. These failures, in my opinion, are due to a combination of factors.

First is the proliferation of imitation right stuff currently being hyped, and the simple fact that those of us with experience don't spend enough time teaching.

My generation was the first to use all of these semi-exotic plumbing paraphernalia and at first we were scared to death of it. So we took the time to learn how it all works, how to assemble it and how to use it. The trouble is that we have known about it for so long that it is now second nature to us and we assume, without thinking about it, that the young ones know as much about it as we do. They don't—simply because we haven't taught them. The really good young ones ask (by definition there can be no such thing as a stupid question). The rest just plow ahead and all of a sudden the plumbing that we have taken for granted for a decade starts to give us trouble. Being human, we blame either the younger generation or the manufacturer. Maybe this will help the young ones.

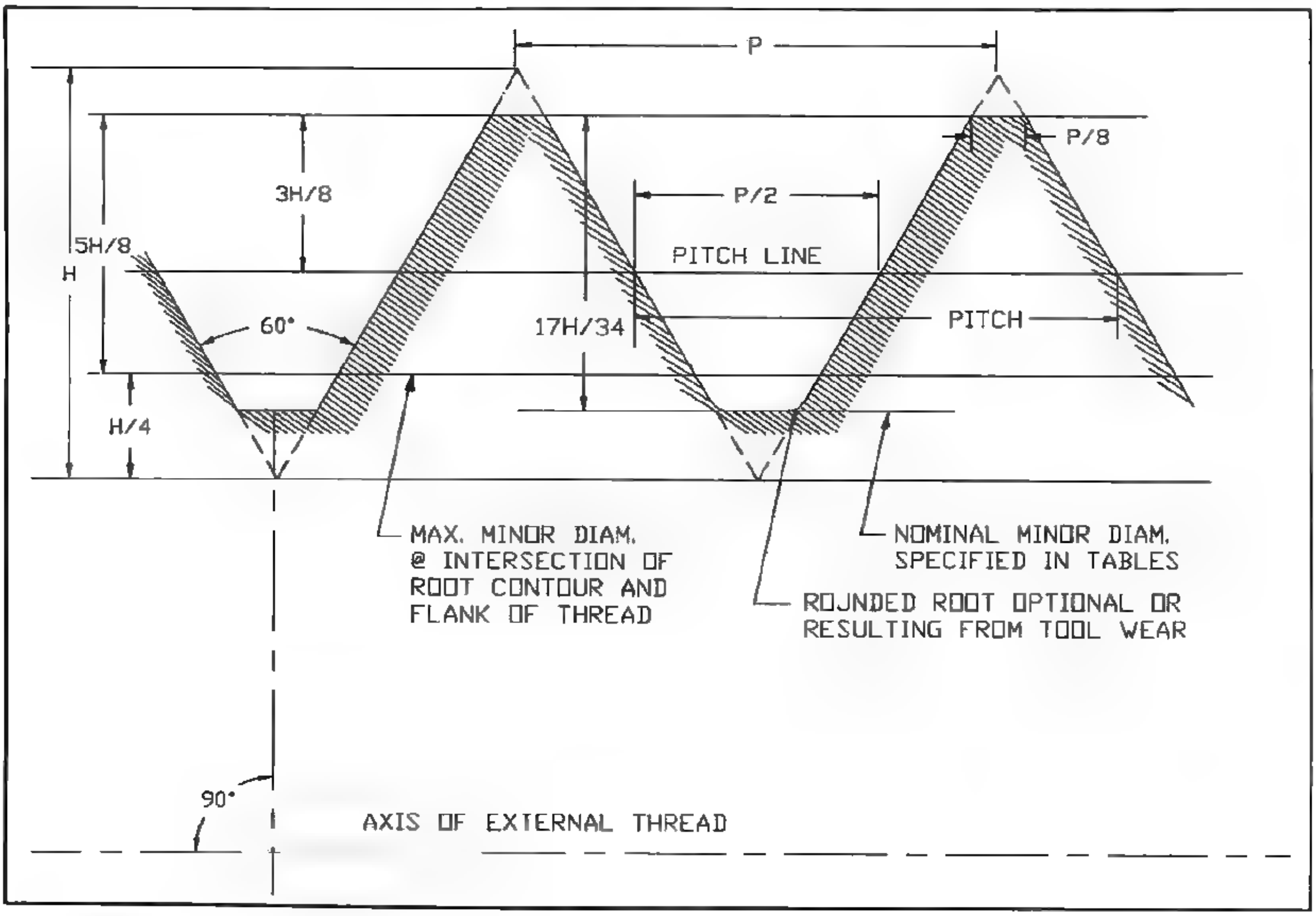
11

Future trends in fastening

In many, if not most, industries, fasteners have historically been the last factor to be considered in a given design—if indeed they are considered at all. Very often the design of joints and the selection of fasteners have been afterthoughts rather than design considerations. Just as the aircraft industry led the way in the development of standardized fasteners in the decades before World War II, the aerospace industry has been the leader in the development of high-strength fastening systems and engineered joints.

The engineers who design our high-performance aircraft and our space vehicles are concerned with both weight and reliability. They know that the greater initial cost of a super-fastener or a completely engineered joint is insignificant when compared to the cost in terms of labor and downtime of replacing a less effective item or system during scheduled maintenance. They also have access to *a lot* of money.

The aerospace industry has created the highly specialized field of fastener engineering. One of the significant aerospace spin-offs that we read about is the high-strength fastener technology developed for exotic applications that is now trickling down to us normal people. The end result is



The Unified Thread Form external thread.

that specialized fastening systems are now available to all of us who have need for them. This spin-off allows us to design and build better, stronger, lighter and longer lasting mechanisms—be they farm tractors, lathes, punch presses, jet skis or race cars. In closing this book, I am going to cite a few examples of what I think the future holds: its problems and possible solutions.

A problem: Strength or fatigue

The basic problem in the joining of materials is not the strength of the fasteners used, or even the strength of the completed joint. You can always make the joint stronger by using stronger materials or larger fasteners. The actual problem is twofold, and involves both weight and fatigue life.

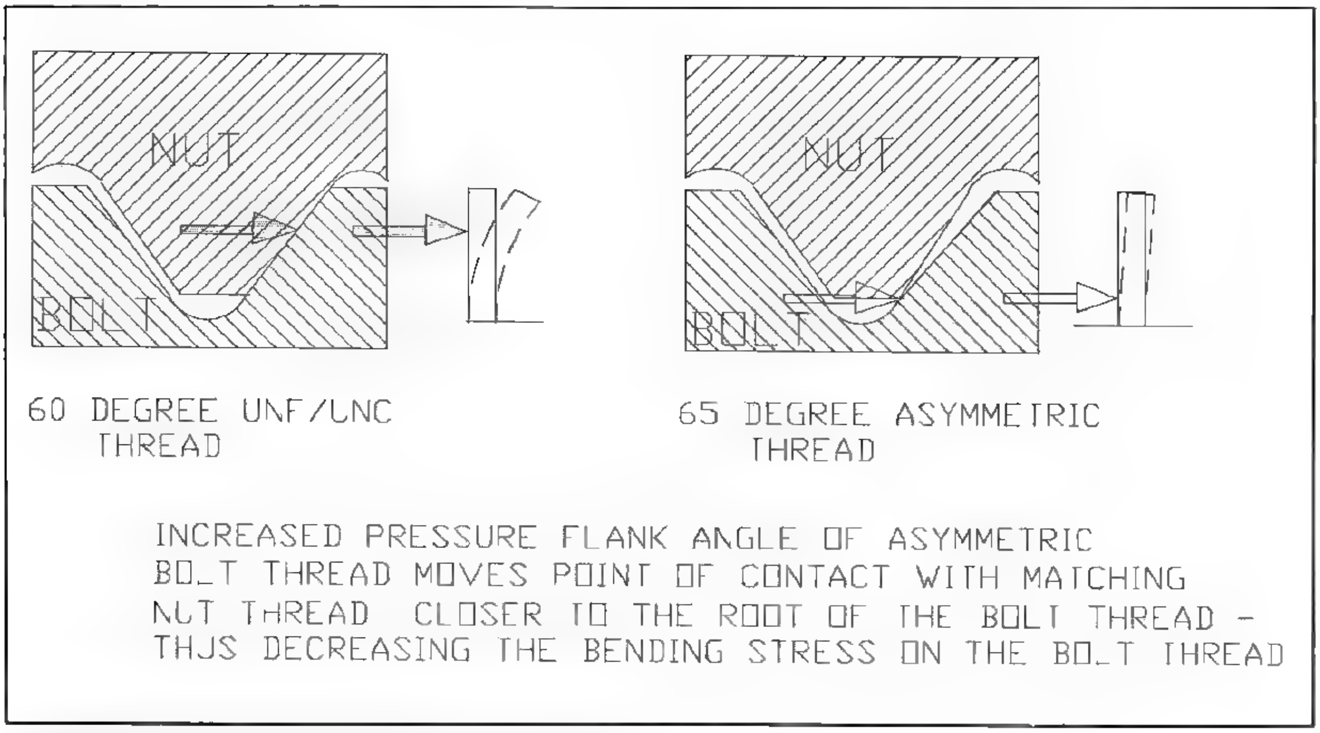
We have seen that the use of a conventional thread in tension concentrates the residual stress on the first few engaged threads of the bolt, thus reducing the fatigue life of the assembly. We have also seen that the use of stronger alloys in the manufacture of fasteners has only a limited effect on fatigue life because the stronger alloys are harder and therefore more brittle and more sensitive to stress raisers and fatigue. The high end of the fastener industry has about run out of practical metallurgical improvements. Recent developments have concentrated on increasing fatigue life of fasteners by sophisticated design of thread forms.

A solution: Asymmetric threads

Once upon a time, not so long ago, the major cause of fatigue failure of threaded fasteners was concentration of stress in the sharp corners of the root of the male thread. The almost universal substitution of rolled threads for cut threads greatly reduced the incidence of stress raisers in the thread root, at least in the high end of the fastener industry. The adoption of the UNR/MIL-S-8879/UNJF radiused root threads in critical aerospace bolts and high-strength commercial bolts virtually did away with the thread root stress raisers.

The next weak link in the fatigue chain proved to be nonuniform distribution of load in and among the engaged threads. With conventional thread forms, stress is concentrated near the roots of the male threads. Further, the residual stress in a tightened fastener is not evenly distributed along the length of the engaged threads, but tends to be greater at the threads closest to the bearing face of the female thread.

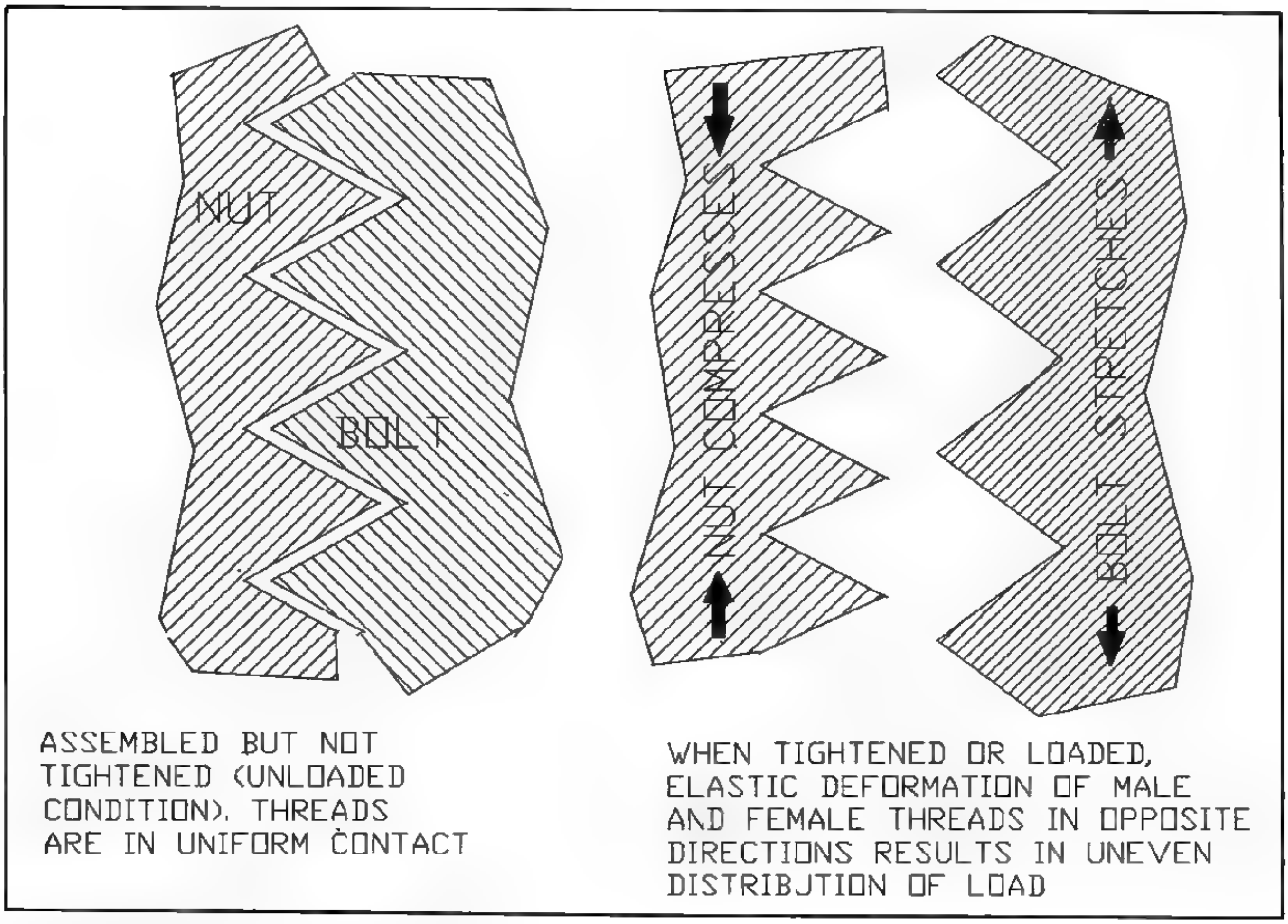
This nonuniform distribution of load is caused by two separate but related phenomena. When the male and female threads are brought into contact and tightened, they act on each other with very high forces (remember the wedge analogy from earlier—we are talking about some fearsome stress here). This stress actually bends the individual ridges of the threads like cantilevered beams. The



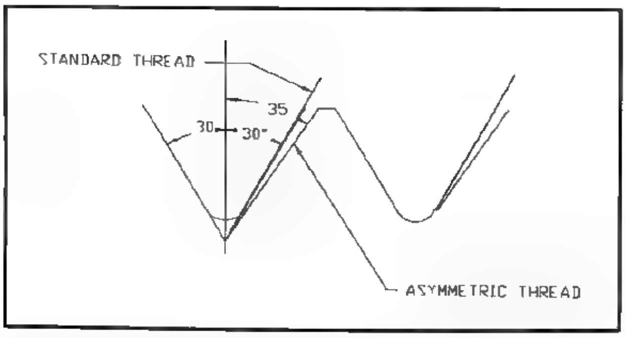
Cross section of both UNF and asymmetric threads in the assembled but untightened condition.

linear deflections produced are quite small, but the bending moment (the distance from the point of contact to the root of the thread) is relatively long. Like a long pry bar, this long moment arm imposes very high stress at the already critical root of the thread.

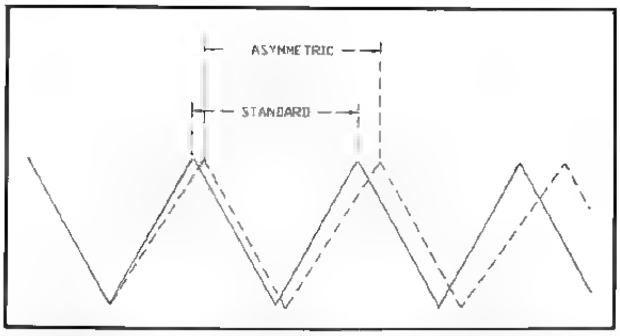
The illustration shows that the 60 degree conventional thread is formed of two symmetric 30 degree flanks. This is true for both male and female threads. Under load, the mating 60 degree threads will theoretically come into contact along a considerable area of their respective pressure flanks. The effective point of load transmission is assumed to be located halfway along this line of contact, relatively high up the pressure flank of the bolt thread. This results in a relatively long moment and a high



Standard 60 degree threads in unloaded and loaded conditions.



Pressure angle of the asymmetric thread compared with that of a standard thread.



The negative lead correction of the SPS asymmetric thread.

stress concentration at the male thread root.

However, manufacturing tolerances and quality control being what they are, this theoretical condition is seldom met in practice. If the pressure angle of the nut thread is slightly greater than that of the bolt thread, the point of contact will be nearer to the crest of the male thread. The result is a still higher concentration of stress at the root of the bolt thread. Conversely, should the nut thread pressure angle be greater, the point of contact will be closer to the root; the moment arm will be shorter and the concentration of stress will be less. With conventional thread forms this desirable condition will occur only by chance.

Under load, a bolt will stretch and a nut will compress. Threads that were evenly mated in the no-load condition are no longer aligned under load. This concentrates the load over relatively few engaged threads and further increases the concentration of stress in the loaded threads. These two weaknesses in the design of conventional thread systems can be largely relieved by the use of the asymmetic thread. In fact, by reducing stress concentrations and providing more uniform load distribution, the asymmetric thread can increase bolt fatigue strength by a minimum of twenty percent over any other thread form to date.

Thread angle

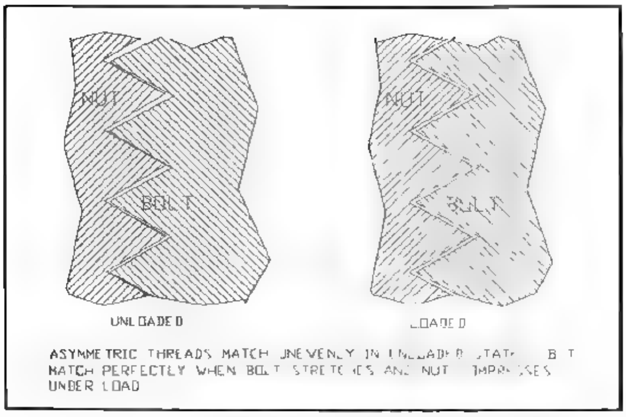
The angle of the pressure flank of the asymmetric thread is 35 degrees instead of the conventional 30 degrees. This forces the point of thread contact downward—as close to the root of the male thread as practical. This ensures that the bending moment is kept at the minimum practical length and so reduces the concentration of stress at the critical male thread root. Of course, the bending moment and the resultant stress at the root of the female thread are correspondingly increased. This is no great worry, as the nut (or the tapped hole) has greater relative mass than the bolt and is not subjected to the complex loads that the bolt sees. Almost all fatigue failures occur in the bolt.

Lead control

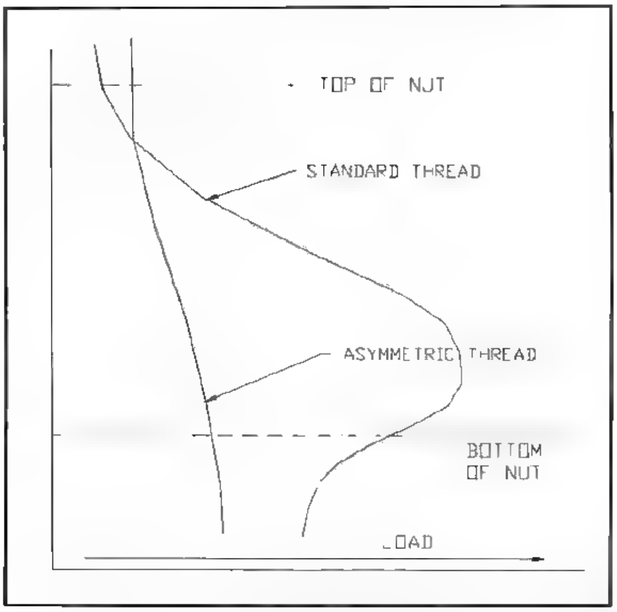
When conventional bolt/nut threads are in contact, but not preloaded, they are assumed to have 100 percent contact. Under load, however, the bolt stretches and the nut compresses. The pitch, or distance between threads, of the bolt increases while that of the female thread decreases. The result is that the installed load is concentrated on the lowermost threads—those closest to the bearing face of the nut. Typically the lower thirty percent of the bolt threads end up carrying about fifty percent of the load. The result is a concentration of stress.

The asymmetric thread compensates for this phenomenon by designing a shorter lead or pitch into the bolt thread. When assembled, but not loaded, consecutive threads of the asymmetric thread do not make uniform contact. However, when tightened to the designed preload, they do. As SPS put it, "the bolt is stretched back to where it should have been in the first place, with all of the threads carrying a nominally equal share of the load."

An asymmetrically threaded bolt can be used with tapped holes, as long as the depth of thread is at least equal to standard nut height. Offhand, I cannot think of any applications where this would not be true. They can be used with standard nuts, but, in order to utilize the full potential of the system, specific companion lock nuts should be used.



Cross section of the SPS asymmetric thread in the assembled but unloaded condition compared with the thread under load.



Comparison of load distribution between standard 60 degree male threads and SPS 5 degree asymmetric threads.

The results

The results are impressive. The fatigue life of existing installations can be greatly extended by substitution of asymmetrically threaded fasteners of the same size and weight as the original.

By designing with either fewer or smaller fasteners, both the mass and the bulk of a given joint can be reduced without shortening the designed fatigue life. As an example, a 7/16–20 asymmetrically threaded bolt at 260,000 psi ultimate tensile strength offers the same fatigue strength as a ½–20 bolt of the same tensile strength and standard threads.

What about the nut?

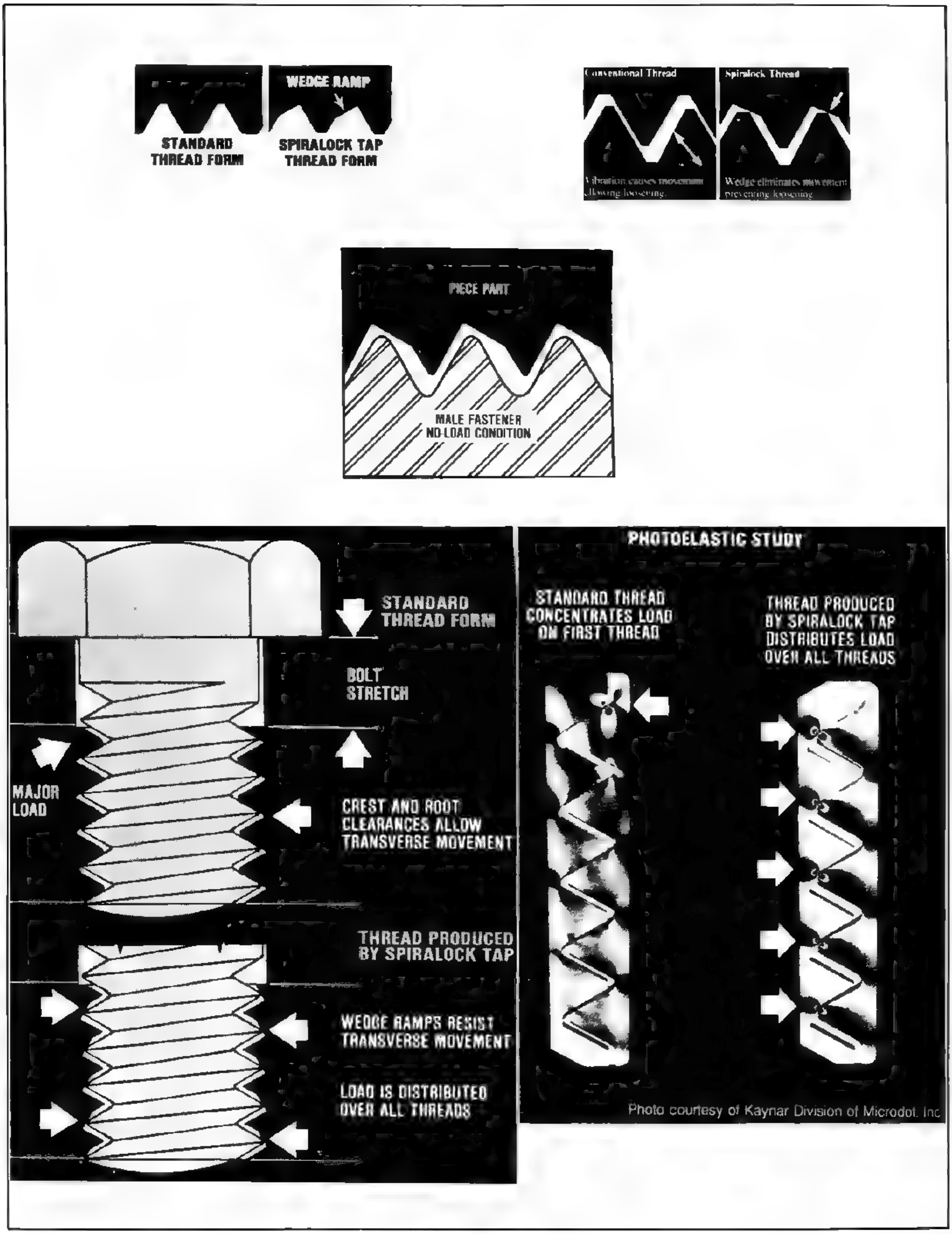
The same problem can also be approached from direction of the female threads. ESNA has an Equa-Stress nut thread that is asymmetric. By varying the thread form, thread angle and pitch, the level of installed stress is kept constant throughout the length of the nut. This relieves the usual stress concentration of the first engaged bolt thread and significantly increases the fatigue life of the assembled fastener.

The higher-strength aerospace threaded fasteners, both bolts and nuts, are now available with

Redistribution of load along the Equa Stress thread form is shown by the stress lines of the photoelastic model. It is this redistribution, which relieves the usual load concentration on the first engaged bolt thread and provides the high fatigue performance.

Photoelastic study of standard thread form graphically illustrates the high stress concentration loads imposed upon the lower threads. This clearly indicates that the majority of the load is placed upon the first engaged thread of the bolt, explaining the high frequency of fracture and failure of the bolt at that point.

The Equa-Stress nut thread form by ESNA.



The Spiralock female thread form.

asymmetric threads as are some of the more exotic rod end bearings.

Variation on the theme

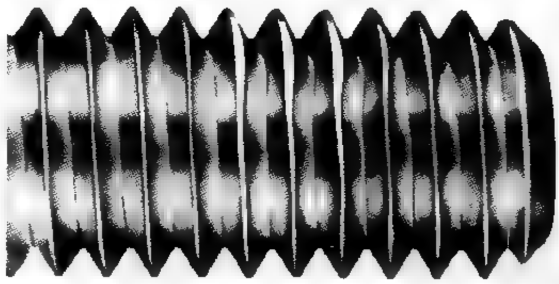
For blind applications, Detroit Tool Industries of Warren, Michigan, has a tap system that cuts a female asymmetric thread. They call system the Spiralock preload locking thread system. It features a two-angled pressure flank. The top of the female thread is angled at 60 degrees instead of the standard 30 to create a wedge ramp. When the bearing face of the bolt bottoms on the work face and the clamping load begins to develop, the crest of the male threads are drawn tightly against these ramps. Elastic deformation of both male and female threads then lock the assembly in pretty much the same way that an all-metal elastic stop nut would. More important, however, is the fact that, like SPS's asymmetric male thread, the Spiralock female thread helps to even out the distribution of installed stress along the length of the engaged male threads. To make things even better, the Spiralock female thread mates with standard male threads so special bolts are not necessary. This is a very significant development.

The Spiralock has survived extensive testing by NASA. The Kaynar division of Microdot is producing nuts, and the Rocketdyne division of Rockwell International uses Spiralock on the space shuttle main engine and on the Peacekeeper missile. In fact, it looks to me like Spiralock's exceptional resistance to vibration will cause it to be adopted very quickly for both tapped holes and for nuts, throughout the automotive, machine tool and aerospace industries.

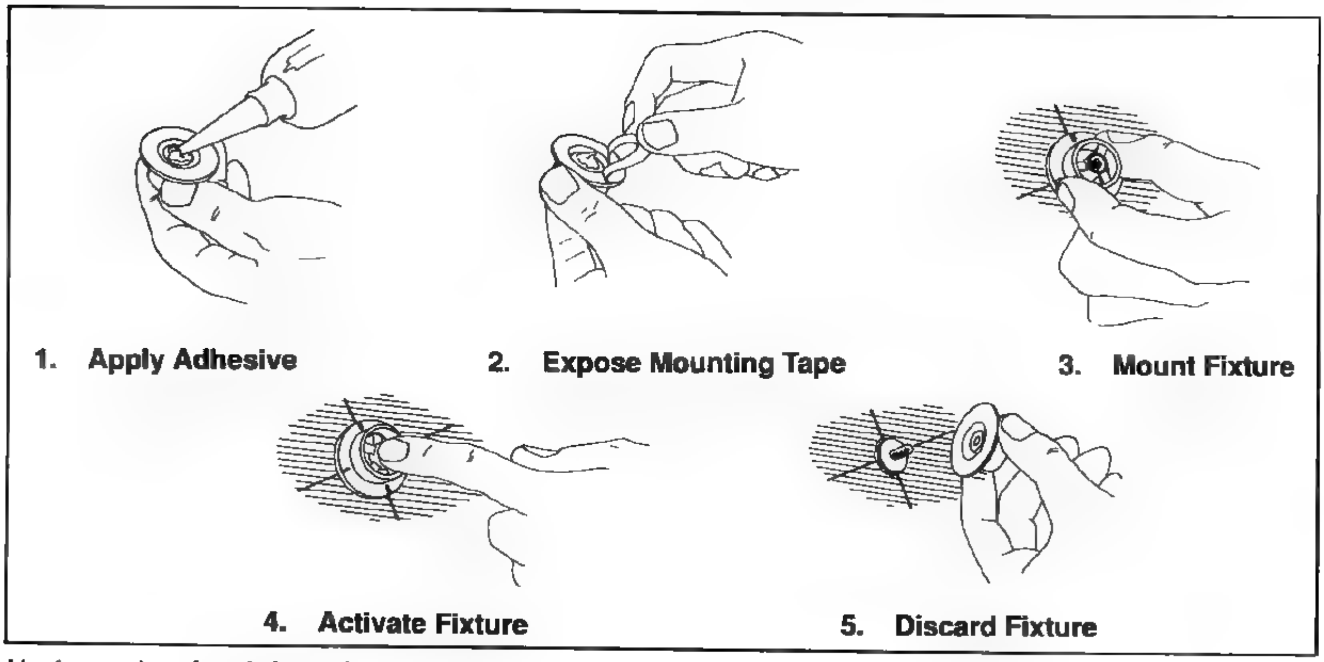
What next?

The immediate future of fastening will see increased use of high-strength fastening "systems" consisting of matched male and female fasteners specifically designed for a given application. Fastening in modern aircraft has evolved from 125,000 psi AN bolts and hand-bucked solid aluminum rivets to 300,000 psi bolts and matching nuts, and to sophisticated blind bolts and positive mechanical locking rivet systems. The automotive industry, in their search for improved fuel efficiency and increased warranty life, is moving in the same direction. The asymmetric male thread and the Spiralock female thread form will doubtless become international standards.

In my opinion, however, the *real* future of fastening is in the field of adhesive bonding. The



Detail of the SPS Tru-Flex fastener. SPS Technologies

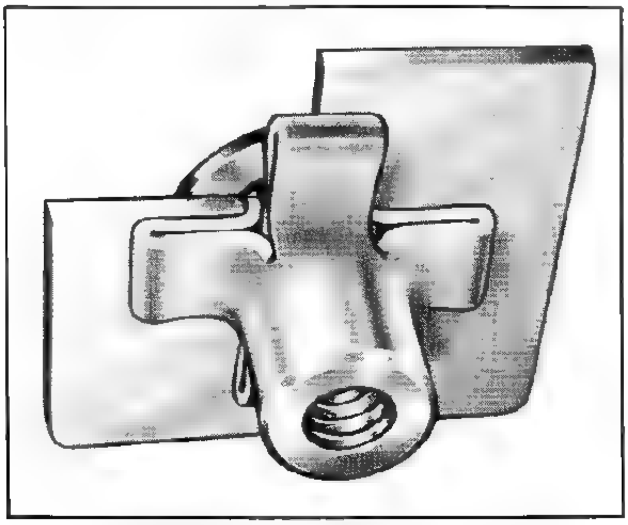


Use instructions for Click Bonds.

automotive industry is beginning to make use of bonding technology. Adhesives are being developed that will bond satisfactorily through light films of contaminants so that extensive and critical surface preparation will no longer be necessary. The decreased labor cost, improved fatigue life and decreased weight of adhesively bonded parts will inevitably lead to greater use of adhesives in general industry and to the development of stronger and less critical compounds. Inevitably the technology will become both available and practical for the individual.

Graphic example

Since I first wrote *Prepare to Win,* more than fifteen years ago, I have been asked time and time again if I really thought that it is worthwhile in this day of technologically advanced everything to write on subjects as basic as nuts, bolts and rivets. My answer has always been that, so long as racing cars and aircraft are crashing because of fastener failures, I sure as hell think it's worthwhile. This attitude has just been confirmed in no uncertain terms by a 150 mph suspension failure originating at the tension failure of an Allen bolt installed in both tension and bending. The driver, my son, escaped unharmed. The car did not. When asked (before the failure) why he was using Allen bolts instead of aircraft hardware, the builder stated that he had been using them for years and, "never had any trouble." If only one designer, constructor or mechanic stops using dubious hardware as a result of reading this book, all of the effort will have been worthwhile.



The B. F. Goodrich Plusnut.

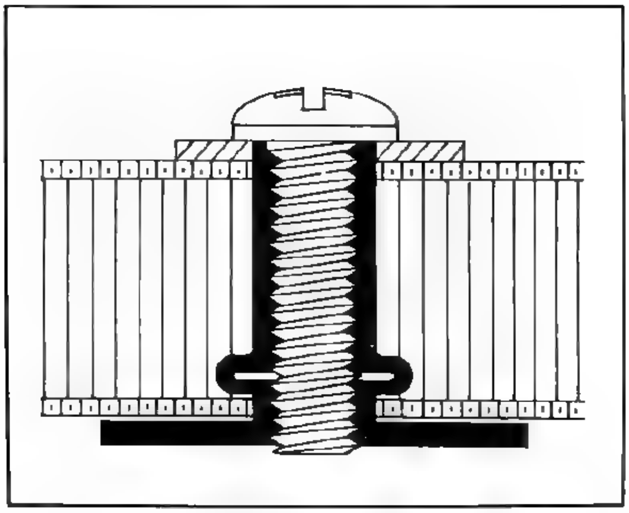
Click Bond fastening systems

It had to happen! Someone had to come up with a practical method of bonding fasteners to metal and/or composite surfaces. Click Bond Inc., a subsidiary of Physical Systems Inc., has done just that. They offer a line of studs, standoffs and nutplates in Metric and UNF threads, and a wide variety of materials. They also offer surface patches.

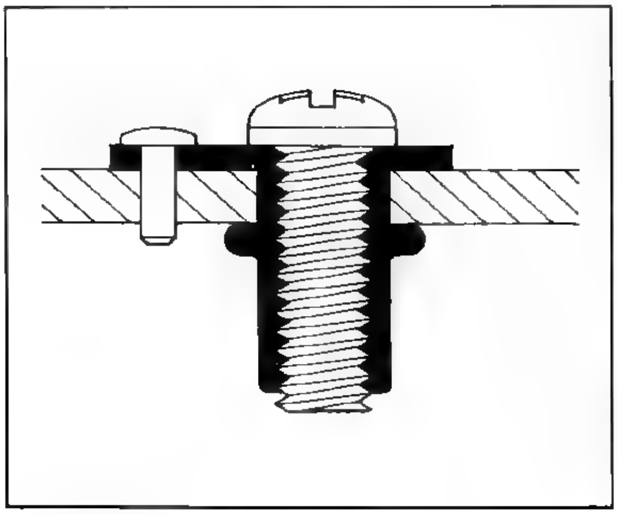
High tech comes to the torque wrench

Those of us outside the aerospace industry depend upon the torque wrench for correct preloading of just about every tension bolt except connecting rod bolts. We discussed why this is a somewhat less than optimum arrangement in chapter 3. What I forgot to say is that even the best torque wrenches (Snap-On and Proto) must be recalibrated at frequent intervals.

Well, the age of the microprocessor has arrived in the world of bolt tensioning. Since they make most of the truly high-tech threaded fasteners in



The Maxihead Rivnut.



The Teardrop Rivnut.

the world, it comes as no surprise that SPS Technologies designs and manufactures virtually all of the automatic bolt tensioning systems for industry. The surprise is that the Aerospace Division of the SPS Technologies is now marketing a manually operated family of microprocessor-controlled bolt tensioning wrenches called the Sensor I family. Available in 125, 250 and 400 lb-ft capacities, these devices have three modes of operation: preset torque, just like the mechanical wrenches we are used to; angle control, which signals when the bolt head has rotated through a preset angular value (after a preset "snug up" value has been reached) thus allowing very precise determination of bolt stretch and JCS-TEL; and a magic mode that senses when the fastener reaches its elastic limit in tension, i.e., yield point. Regardless of what I said in chapter 3, it is now possible for each of us to accurately (if not cheaply) measure bolt installed tension/preload.

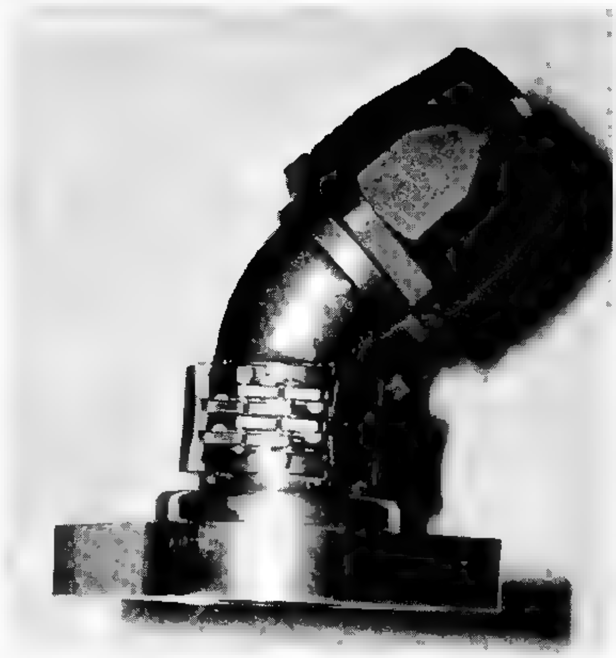
Rivnuts for soft materials and honeycomb

Just as Cherry has developed rivets for blind fastening of soft materials and honeycomb panels, B. F. Goodrich has developed a series of Rivnut derivatives to provide permanent threads in the same types of materials and panels. The Plusnut features a large-diameter finished head and upsets into a four-petaled shop-formed head with increased bearing area for use with soft materials: plastics, fiberglass, Kevlar and so on. The Maxihead Rivnut is designed for use with honeycomb panels.

Also new from B. F. Goodrich is the Teardrop Rivnut which makes damned sure that you are not going to twist the thing out of its hole. Most BFG Rivnuts are now available with the Unilock deformed thread locking feature.

AVK Industrial Products

New to me, if not to the rest of the world, is AVK Industrial Products, "manufacturers of threaded inserts for industry." These people are a division of Aviabank Manufacturing. Their catalog lists just about every type of blind threaded insert I have ever heard of and a couple that I have not seen before. They combine Rivnut types and Nut-Sert types in one source.



Earl's Easy-Loc coupling.

More trick stuff from Earl's

For some years now we have been using Wiggins Couplings on the cold side of our turbocharger installations. They offer a truly fast and foolproof method of attaching lengths of rigid tubing in such a way that is vibration resistant and will allow limited misalignment between adjacent lengths of tubing. They are expensive but worth it in critical applications where speed of replacement and misalignment capacity are critical. One of the nice features of this device is that it is instantly inspectable for proper assembly—if the clamp is in place, the coupling is sealed—there is no tightening involved so it cannot be left loose.

Earl's Performance Products has now developed a line of hose ends and adaptors which utilize the Wiggins couplings in new configurations. Originally intended to use on engine water lines to allow fast and foolproof engine changes, the racers are coming up with new uses daily. I use them for water lines both at the radiator and at the engine. The hose end style is called Easy-Loc and the hard line coupler is called Wig-O-seal.

Appendices

Recommended reading

I see no sense in my covering ground that others have covered well. Those of you who are interested in going further into any of the subjects that I have discussed (all of you, I hope) are hereby urged to obtain the following resources.

Light Airplane Construction for the Amateur Builder, by L. Pazmany, available from L. Pazmany, P.O. Box 80051, San Diego, CA 92138. This is *the best* book of its type ever written. No one who intends to do any sheet-metal fabrication or fiberglass work can afford to be without it. Pazmany has also published the construction manual for his Pazmany PL–4 homebuilt airplane, an exceptionally clever design. Not all of the information in the construction manual is contained in the guide. The construction manual is worth buying just to see how simple, clever and complete the man's work is.

Pazmany has just published another book, *Landing Gear Design for Light Aircraft.* I have no intention of designing landing gear for an aircraft, light or heavy—but I have learned a lot from this highly specialized book. I strongly recommend that anyone concerned with either vehicle dynamics or vehicle construction read it—both for its content and for Pazmany's approach to engineering.

Low Power Laminar Aircraft Structures, by Alex Strojnik, available from Aircraft Spruce and Specialty Company. Another milestone book on the techniques of building just about anything correctly. Other books include *Laminar Aircraft Technologies* and *Laminar Aircraft Design.* He thinks and writes well and it is always good for us to learn how other clever people approach problems.

Sport Plane Construction Techniques, by Tony Bingelis. Available from the author at 8509 Greenflint Lane, Austin, TX 78759, from Aircraft Spruce or from the Experimental Aircraft Association. As nearly as I can figure, Tony Bingelis is the Carroll Smith of the homebuilt aircraft world. I hope that my writing about racing cars is as clear as his about aircraft. This book covers just about every facet of building your own aircraft. Naturally there is lots of information about hardware.

I have previously mentioned the manuals published by the Experimental Aircraft Association—upon whom blessings be. This time I am going to be specific. You should own: *Sheet Metal,* Volume One and Volume Two; *Tips on Fatigue; Building the Metal Airplane;* and *Aircraft Maintenance Manual.* They are available at nominal cost from the Experimental Aircraft Association, Wittman Airfield, Oshkosh, WI 54903–2591. It is well worth joining the association just to receive their monthly magazine, *Sport Aviation.* (Anyone who is in the Elkhart Lake area or at the Milwaukee Mile and does not visit their museum at Oshkosh is missing something really worthwhile.)

The Hitchcock Publishing Company publishes an annual entitled *19XX Assembly Technology Buyers Guide* which is an illustrated explanation of the fastener industry, an excellent source book. Their address is:

The Hitchcock Publishing Company
Hitchcock Building
Wheeling, IL 60188

Back around the dawn of the age of aviation a very clever man named Wendell Fletcher founded Fletcher Aviation in El Monte, California. When the industry underwent its overnight expansion of wartime status in 1941, Fletcher wrote and published his *Standard Aircraft Workers' Manual* as a training aid for the tens of thousands of brand new aircraft fabricators and assemblers. It worked in 1941 and, in its fifteenth revised edition, it works today. There is no better introduction to the basics of drafting, assembly, wiring and fastening. It should be in every toolbox. You might get lucky and find it at your general aviation store. If not, it is available from these two sources:

Sargent-Fletcher Aviation
9400 Fair Dr.
El Monte, CA 91731

Fletcher Aircraft Company
RFD #4, Box 25
Sedalia, MO 65301

Sources

The subject of high-performance hardware is so large and there are so many different products out there that, even in a book of this size, we can only skim the surface of what is available. By the same token, the number of catalogs available approaches the infinite. The following list is my idea of the best sources that are available in each field.

Rod end and spherical bearings

Baker Precision Bearings
2865 Gundry Ave.
Long Beach, CA 90806

The prime distributor of NMB rod end and spherical bearings to the racing trade. They also handle most other lines. Paul Baker probably knows more about rod end and spherical bearings than anyone in the business—he is able to match the bearing to the job. They are also into very high quality synthetic lubricants. Their NEO gear lubes and greases are the best that I know of and are used by many Formula One and CART teams.

Bolts

SPS Technologies
Aerospace and Industrial Products Division
Highland Ave.
Jenkintown, PA 10946

"SPS Technologies Reference Guide to Bolts and Screws," Section I is a pictorial listing of just about every AN, MS and NAS bolt and machine screw. They are classified as tension or shear units and listed in decreasing order of strength. I use it as a general reference to what is available and to identify what I find in surplus stores. Section III is a detailed catalog, with specifications of the bolts listed in Section I.

SPS publishes several different catalogs, spec sheets and brochures. Your chances of obtaining any of them from the source are remote. Your chances of obtaining the obsolete ones from a fastener distributor are very good.

Automotive Racing Products (ARP)
8565 Canoga Ave.
Canoga Park, CA 91304

Manufactures and markets a comprehensive line of threaded fasteners for high-performance engines.

Metric bolts

Metric and Multistandard Components Corporation
120 Old Saw Mill River Rd.
Hawthorne, NY 10532

Catalog No. 4011 lists just about every metric and British industrial known to man, complete with dimensional and strength specifications. They even include tools, taps, dies, gauges and pneumatic fittings.

Global Metrics Inc.
519-J Marine View Ave.
Belmont, CA 94002

Another outstanding metric industrial hardware catalog.

Metric Specialties Inc.
410 S. Varney St.
Burbank, CA 90502

Metric fasteners.

Nuts

Standard Pressed Steel Company
Jenkintown, PA 19046

"SPS Technologies Reference Guide to Self-Locking Nuts" (1985), and "SPS Self-Locking Nuts" (Form 3013 366–25M–SPS).

ESNA Division, Amerace
2330 Vauxhall Rd.
Union, NJ 07083

ESNA Aerospace Catalog 676, "Elastic Stop Nuts for Commercial and Military Aircraft Aerospace Applications."

Kaynar Mfg. Co. Inc.
Kayloc Division
P.O. Box 3001
Fullerton, CA 92634

Kayloc modern lightweight self-locking nuts catalog.

AN, MS, NAS fasteners and specialized high-strength fasteners

ARP (Automotive Racing Products)
8565 Canoga Ave.
Canoga Park, CA 91304

Manufacturers of specialized high-strength, fatigue-resistant threaded fasteners for racing engines, ARP products are the only first-class, no-hype aftermarket engine fasteners that I know of. Their catalog is a goldmine of information.

Coast Fabrication Inc.
17712 Metzier Ln.
Huntington Beach, CA

These guys stock a complete line of AN, MS and NAS fasteners. They know what they are talking about and both the price and the service are good.

Dixon Steel Fabrication
RD 1
Pleasant Mount, PA 18453

Dixon markets a complete line of rod end bushings and spacers, taper spacers, captive washers, AN washers, jam nuts, elastic stop nuts and a limited line of AN bolts. The spacers are one of those things that it just isn't worth making yourself.

Threaded inserts

The B.F. Goodrich Company
Aerospace and Defense Division
250 N. Cleveland-Massillon Rd.
Akron, OH 44313

Rivnut-Plusnut design guide.

AVK Industrial Products
25323 Rye Canyon Rd.
Valencia, CA 91355–1271

These people have endless configurations of Rivnut-type threaded inserts, ball lock pins, panel fasteners and the like.

Panel fasteners and latches

The Rexnord Corporation
Specialty Fastener Division
3000 W. Lomita Blvd.
Torrance, CA 90505

"Camloc ¼-turn fasteners," catalog #1000 and "Camloc Latches," catalog #1100.

Dzus Fastener Co. Inc.
425 Union Blvd.
West Islip, NY 11795

The Dzus general catalog.

Dimco-Gray Company
8200 S. Suburban Rd.
Centerville, OH 45459

"Dimco-Gray Snapslide Fasteners," catalog SSF–84.

General hardware

Aircraft Spruce and Specialty Company
201 W. Truslow Ave.
Fullerton, CA 92632

Suppliers of composite materials and parts, aircraft specification hardware, supplies and tools for the homebuilder of aircraft. Their catalog is one of the *great* ones.

California Aero Supply
13840 Paramount Blvd.
Paramount, CA 90723

Cal Aero does *not* have a catalog. What they are is the last of your basic old-fashioned surplus fastener houses. You have to do some hunting—but the base price for steel surplus is under a dollar a pound. They usually have a pretty good selection of AN bolts, washers, plate nuts and so on.

The Dillsburg Aeroplane Works
Dillsburg, PA 17019

Owner Charlie Vogelsang stocks a complete line of AN hardware, tubing and sheet metal.

Rivets

Pop Fasteners Division
Emhart Fastener Group
510 River Rd.
Shelton, CT 06484

Blind riveting catalog and engineering handbook.

Avdell Corporation
50 Lackawana Ave.
Parsippany, NJ 07054

"The Avex blind breakstem rivet system and technical supplements," "Avdell engineered assembly systems," "Avdell blind structural breakstem fasteners" and "Avdell aerospace fastener data."

Pop and Avdell literature is available from your local rivet distributor in the yellow pages.

Huck Manufacturing Company
6 Thomas
Irvine, CA 92714

Cherry Division of Textron, Inc.
1224 E. Warner Ave.
Santa Ana, CA 92707

Commercial fastening systems catalog.

Cables

Cablecraft Inc.
2011 S. Mildred St.
Tacoma, WA

ACCO Industries Inc, Cable Controls Group
220 Industrial Dr.
Milan, TN 38358

Morse Controls, INCOM International Inc.
75 Clinton St.
Hudson, OH 44236

Plumbing

Earl's Performance Products
825 E. Sepulveda
Carson, CA 90745

Earl's Performance Products manufactures Swivel Seal, Speedflex and Auto-Fit hose ends, Perform-O-Flex and Speed Seal hose as well Temp-A-Cure oil coolers. They stock a complete line of aircraft and racing hardware and support equipment, and publish an excellent catalog. Earl's products are available nationwide through selected distributors and dealers. *The* plumbing house for racer.

Aeroquip Corporation
300 S. East Ave.
Jackson, MI 49203

Aeroquip is the biggest name in the flexible hose and reuseable hose end business. Their products are second to none in quality and reliability. Aeroquip is represented by a nationwide chain of industrial distributors. Most of them don't know much about high-performance plumbing parts and don't stock them. Given enough time and money, their aircraft distributor will have what you need or can get it for you.

Torino Motor Racing Ltd.
1350–M W. Collins
Orange, CA 92668

Mike Torino is one of the leading distributors of Earl's Performance Products. They are also the only source that I know of for a lot of specialized hardware: AN to BSP adaptors, Copaslip antiseize, Hylomar gasket compound, oddball AN fittings, plate nuts with replaceable elastic stop nuts—the kind of stuff that only hardcore racers need or even understand. Mike or his right-hand man, George Shippers, can bail you out of almost any kind of plumbing related trouble that you can get yourself into.

Dixon Steel Fabrication
RD#1
Pleasant Mount, PA 18453

Dixon markets a complete line of rod end bushings and spacers, taper spacers, captive washers, AN washers, jam nuts, elastic stop nuts and a limited line of AN bolts. The spacers are one of those things that it just isn't worth making yourself.

The Eastwood Company
580 Lancaster Ave.
Malvern, PA 19355

They specialize in restoration tools for classic and antique cars. They have a number of items that are hard to find elsewhere: dimpling pliers, joggling pliers, shot bags and so on.

Miscellaneous

The Long Lock Corporation
4101 Redwood Ave.
Los Angeles, CA 90066

Pressure activated adhesive coating for threads.

Nylock Fastener Corporation
800 W. University Dr.
Suite E
Rochester, MI 48063

Pressure activated adhesive coating for threads.

Accurate Automatic Parts Inc.
2885 S. 163rd St.
New Berlin, WI 53151

Locking studs.

Tuck Jones Engineering
P.O. Box 331
Gualala, CA 95445

Drill jig for drilling holes in bolts for safety wiring.

Stage 8 Fasteners
1537A Fourth St.
San Rafael, CA 94901

Antirotational devices.

Detroit Tool Industries
2200 Eleven Mile Rd.
Warren, MI 48091

Spiralock preload locking thread system.

Click Bond Inc.
2151 Lockheed Way
Carson City, NV 89706

Click Bond fasteners.

Lavender Fastener
884 W. 18th St.
Costa Mesa, CA 92627

Industrial and aerospace rivets and threaded fasteners.

Dillsburg Aeroplane Works
Dillsburg, PA 17019

Outstanding East and Midwest source for AN and MS hardware, tube and sheet stock. Next day UPS service.

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